THE INTERNAL-COMBUSTION ENGINE

2

Volume I

SLOW-SPEED ENGINES

BY

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PREFACE

This volume—the first of a two-volume work on the subject of Internal-combustion Engines—deals more particularly with what may be described as the slow-speed type of internal-combustion engine. Unlike the high-speed engine which, under the stimulus of the Great War, has developed by leaps and bounds, the slow-speed type has undergone comparatively little change, so that although the material of this volume was prepared before the War, and includes the description of specific examples of slow-speed engines some of which are no longer being produced, the general statements and descriptions hold good to-day. No doubt the great advance which has been made in both the theoretical and mechanical development of the high-speed engine will influence the design of the slow-speed type in the future, but as yet this influence has not become apparent to any appreciable extent.

The Author hopes shortly to be able to complete the second volume, dealing with high-speed engines, and embodying as far as possible the results of recent research and development.

In conclusion, the Author would like to express his great indebtedness to the late Professor Bertram Hopkinson for having, by his inspiring teaching, awakened and fostered in him a keen interest in the internal-combustion engine in all its phases. His thanks are due also to Sir Dugald Clerk for his persistent encouragement and sound advice; to his friend and colleague Mr. H. A. Hetherington for much valuable assistance; to Mr. Alan E. L. Chorlton for his ready help; and last, but not least, to all those manufacturers who have so freely and liberally supplied him with material for descriptions, together with drawings and photographs of actual engines.

H. R. R.

Suppose also that the temperature of the air before compression were 519° F. absolute, then, after being suddenly compressed to one-third of its original volume, the final temperature of the air will be

$$
\begin{aligned}
T_1 &= T \times 3^{(\gamma-1)},\\
\text{or } T_1 &= 519 \times 3^{(1\cdot 4-1)}\\
&= 519 \times 3^{0\cdot 4}\\
\text{Now, } \log 3^{0\cdot 4} &= \log 3 \times 0\cdot 4 = 0\cdot 477 \times 0\cdot 4\\
&= 0\cdot 1908 = \log 1\cdot 55,\\
\therefore T_1 &= 519 \times 1\cdot 55\\
&= 804\cdot 5^\circ \text{ F. absolute}\\
&= 804\cdot 5 - 459, \text{ or } 345\cdot 5^\circ \text{ F. on the ordinary Fahrenheit scale.}
\end{aligned}
$$

Conversely, if air at an absolute temperature of 804·5° F., and an absolute pressure of 68·5 lb. per square inch, be suddenly expanded to three times its volume, the final temperature will be 519° F. absolute, and the final pressure 14·7 lb. per square inch absolute.

In the cylinder of an internal-combustion engine the compression and expansion of the working fluid is so rapid, that the pressure-volume relation is much more nearly adiabatic than isothermal.

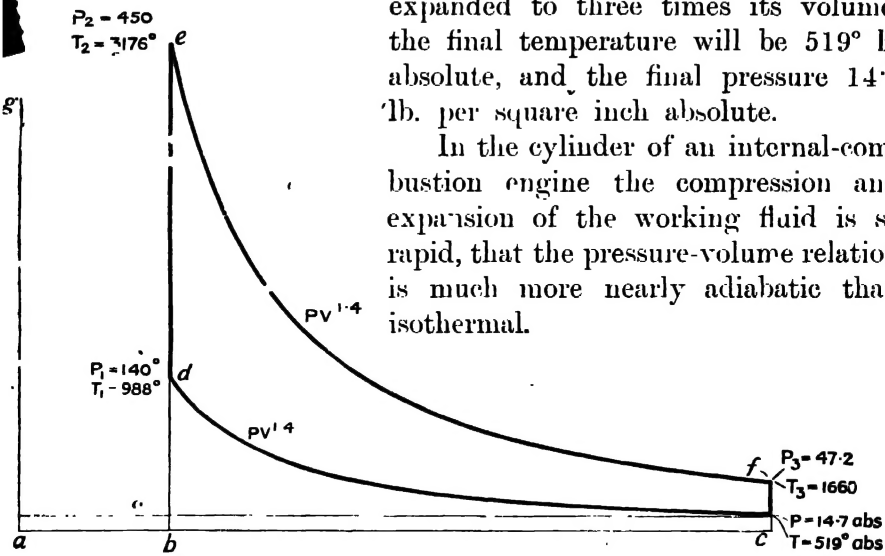


Fig. 3

Indicator Diagram of Constant-volume or Explosion Engine.—Fig. 3 represents an ideal indicator diagram for a four-cycle engine of the explosion type, that is, of the type in which both fuel and air are present in the cylinder during the compression stroke, and in which combustion takes place while the piston is passing its inner dead centre. It might equally well apply to a two-cycle engine of the same type, if the last 20 per cent of the stroke be neglected, for the thermal conditions are almost identical

in both cases. Referring to fig. 3, the horizontal line *b c* represents the travel or stroke of the piston, and the distance *a b* the clearance or compression space into which the air is compressed, or, in other words, is equal to the distance between the piston and the cylinder cover when the former is at its inner dead centre. It is assumed in this case that the compression space represents 20 per cent of the total volume, i.e. that the distance *a b* is one-fifth of the total distance *a c*, which is a very common proportion for small engines using rich fuels, such as illuminating gas or petrol vapour. The vertical line *a g* denotes the pressure in pounds per square inch. Commencing with the piston at the point *c*, we will suppose that the cylinder is completely filled with a perfect and homogeneous mixture of gas and air at atmospheric pressure and temperature. The atmospheric pressure may be taken as 14·7 lb. per square inch absolute, and the temperature at 60° F. or 519° absolute. As the piston travels from *c* to *b* the gas is compressed adiabatically, both pressure and temperature rising until the point *d* is reached, when the working fluid has been compressed into a space 20 per cent of its original volume. Let V represent the total volume and V_1 the compression space or volume when the piston is at *b*, also let P, P_1, P_2, P_3, and T, T_1, T_2, T_3 represent the pressures and temperatures respectively at the points *c*, *d*, *e*, *f*. At the point *c*, P = 14·7 lb. per square inch absolute and T = 519° F. absolute. The pressure P_1 at the point *d* is given by the equation $P_1 = P\left(\frac{V}{V_1}\right)^{\gamma}$. Resolving the equation, $P_1 = 14{\cdot}7\left(\frac{5}{1}\right)^{1{\cdot}4} = 140$ lb. per square inch absolute, or 125·3 lb. per square inch above atmosphere.

It is next required to find the temperature at the point *d*, which is given by the equation $\frac{T_1}{T} = \left(\frac{V}{V_1}\right)^{\gamma-1}$. Now T = 519° absolute and $\left(\frac{V}{V_1}\right) = 5$. Therefore $T_1 = 519 \times 5^{0{\cdot}4}$, whence $T_1 = 988°$ absolute or 529° F. Having now arrived at the temperature and pressure at the point *d* at the end of the compression stroke immediately before ignition, it remains to find the temperature and pressure at the points *e* and *f*, *e* being the point where combustion is complete, but before the piston has started on its outward stroke, and *f* the point at which the heated gases have expanded to their original volume. At this point the exhaust valve or ports are opened and the products of combustion released. The tempera-

ture, and therefore the pressure, at the point *e* will depend upon the quantity and the heating value of the gas mixed with the air, the richer the mixture the higher the temperature and pressure, until a point is reached when the proportion of air present in the cylinder is insufficient to combine with the whole of the gas. In practice the strongest mixtures are not necessarily employed, a slight excess of air being often found desirable; with an average strong mixture of gas and air the pressure at the point *e* will rise to about 450 lb. per square inch, the temperature in this case being

$$\frac{P_2}{P_1} = \frac{T_2}{T_1},$$

$$\frac{450}{140} = \frac{T_2}{988},$$

$$\text{whence } T_2 = 3176^\circ \text{ F. absolute.}$$

Finally the air and gas now expand adiabatically to the point *f*; the temperature T_3 at *f* may be found from the equation

$$T_3 = T_2 \div 5^{\gamma-1},$$

$$= \frac{3176}{1 \cdot 904}$$

$$= 1668^\circ \text{ F. absolute.}$$

The absolute pressure corresponding to this temperature is 47·2 lb. per square inch.

It should be carefully observed that the efficiency, temperatures, and pressure we have so far calculated are ideal ones. The ideal conditions under which the exchanges of mechanical work and heat take place should be carefully noticed.

Influence of Compression on Efficiency.—The next point to be investigated is the thermal efficiency that may be expected from such an engine. The term efficiency is often very loosely applied, but throughout this book it may be taken to mean the ratio between the total heat accounted for as useful work done on the piston and the total heat supplied to the engine, unless expressly stated otherwise.

It has already been shown that if H represents the total amount of heat supplied to the engine at each cycle, and H_1 the amount of

heat discharged by the engine after doing useful work, then the efficiency of the engine is given by the formula

$$E = \frac{H - H_1}{H}.$$

The heat is supplied in this case at constant volume

$$H = K_v(T_2 - T_1).$$

The heat is discharged also at constant volume, and is equal to

$$H = K_v(T_3 - T).$$

The efficiency (E) is therefore

$$E = \frac{K_v(T_2 - T_1) - K_v(T_3 - T)}{K_v(T_2 - T_1)},$$

$$E = 1 - \frac{T_3 - T}{T_2 - T_1}$$

$$= 1 - \frac{1668 - 519}{3176 - 988}$$

$$= 1 - 0{\cdot}515 \text{ approximately}$$

$$= 0{\cdot}475, \text{ or, as it is generally expressed, } 47{\cdot}5 \text{ per cent.}$$

Since the heat is both supplied and discharged at constant volume, and since both the compression and expansion are adiabatic, it follows that the efficiency is dependent solely upon the ratio of compression, and is independent of either the maximum or the initial temperature of the gases. When once the ratio of the clearance volume to the total volume is known, the theoretical efficiency can be arrived at from the simple formula

$$E = 1 - \left(\frac{V_1}{V}\right)^{\gamma-1} \text{ or } E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}.$$

This is generally known as the air standard efficiency.

Efficiency of Diesel or Constant-pressure Type Engine.—Taking next engines of the second type, in which air alone is compressed in the cylinder and the fuel is admitted during the expansion stroke. In this case fuel is injected at such a rate that the pressure remains constant, due to the heating of the air by the combustion of the fuel, during the first part of the expansion stroke. At a certain point in the stroke, the supply of fuel is cut off, and expansion proceeds until the end of the stroke. Figs. 4, 5, and 6 are ideal diagrams for such an engine, and are

equally applicable to engines operating on either the four- or two-stroke cycle. For reasons which will be explained later, it is possible to use a very much higher compression pressure with engines of this type, and, for purpose of comparison, the compression ratio in this case has been taken as 12 : 1. Starting with the piston at the point *c*, it is assumed that the cylinder is filled with pure dry air at an absolute pressure (P) of 14·7 lb. per square inch and at an absolute temperature (T) of 519° F. As the piston travels from *c* to *d* the air will be compressed, until at the point *d* the pressure P_1 will be

$$P_1 = 14{\cdot}7 \times \left(\frac{12}{1}\right)^{1{\cdot}4}$$
$$= 14{\cdot}7 \times 32{\cdot}4$$
$$= 476 \text{ lb. per square inch absolute,}$$

and the temperature T_1 at the point *d* may be found from the equation

$$T_1 = T \times (12)^{\gamma-1}$$
$$= 519 \times 12^{0{\cdot}4}$$
$$= 519 \times 2{\cdot}70$$
$$= 1401° \text{ absolute.}$$

At the point *d* fuel is injected into the highly compressed and heated air, where it immediately ignites, and the admission is continued at such a rate that a constant pressure is maintained during the outward stroke of the piston, until the point *e* is reached, when the supply

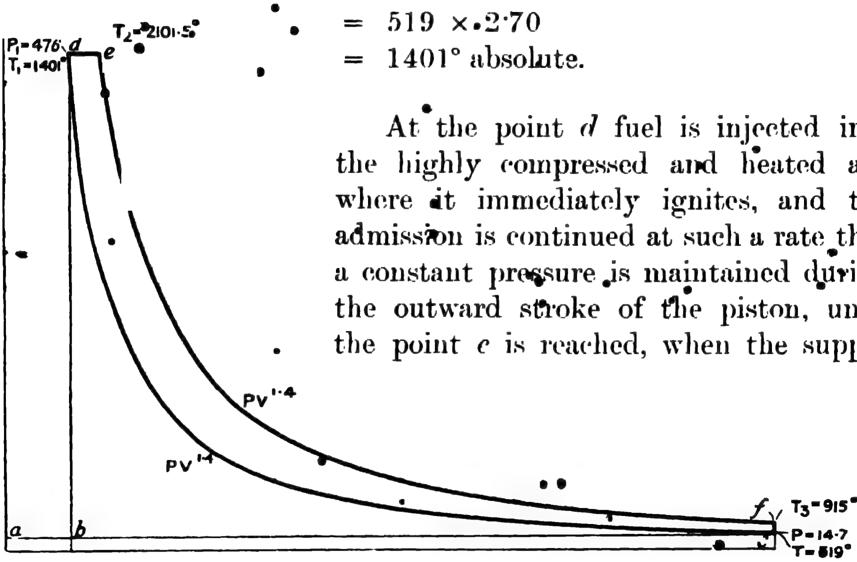


Fig. 4

of fuel is cut off. The efficiency is dependent in this case upon the maximum temperature as well as the compression ratio, and it will be well to take, say, three examples in which the supply of fuel is cut off at (fig. 4) $\frac{1}{22}$nd of the stroke, (fig. 5) $\frac{1}{11}$th of the stroke, and (fig. 6) $\frac{1}{5{\cdot}5}$th of the stroke, or $\frac{1}{8}$th, $\frac{1}{6}$th; and $\frac{1}{4}$th of the total

volume including the clearance space. Taking first example (fig. 4), when the fuel is cut off at $\frac{1}{22}$nd of the stroke, which is about the normal condition when the engine is running on light loads. The volume at e is $\frac{1}{12} + \frac{1}{24} = \frac{1}{8}$th of the total, and the temperature T_2 at the point e is therefore $\frac{12}{8} \times 1401°$ F. $= 2101·5°$ F. absolute. The temperature T_3 at the point f, when the gases have expanded down to their original volume, may be found from the equation

$$
\begin{aligned}
T_3 &= T_2 \div 8^{\gamma-1} \\
&= 2101·5 \div 2·297 \\
&= 915° \text{ F. absolute.}
\end{aligned}
$$

In constant-pressure engines heat is added at constant pressure and discharged at constant volume, so that the formula for arriving at the thermal efficiency is not quite so simple as in the explosion type of engine.

The heat added is $H = K_p(T_2 - T_1)$,

and the heat discharged is $H_1 = K_v(T_3 - T)$.

The efficiency is therefore $E = 1 - \dfrac{K_v(T_3 - T)}{K_p(T_2 - T_1)}$.

Now $\dfrac{K_v}{K_p} = \dfrac{1}{\gamma}$,

so that the efficiency

$$
\begin{aligned}
E &= 1 - \frac{T_3 - T}{\gamma(T_2 - T_1)} \\
&= 1 - \frac{915 - 519}{1·4(2101·5 - 1401)} \\
&= 1 - \frac{396}{1·4(700·5)} \\
&= 1 - \frac{396}{980·7} \\
&= 1 - 0·403 \\
&= 0·593 \text{ or } 59·3 \text{ per cent.}
\end{aligned}
$$

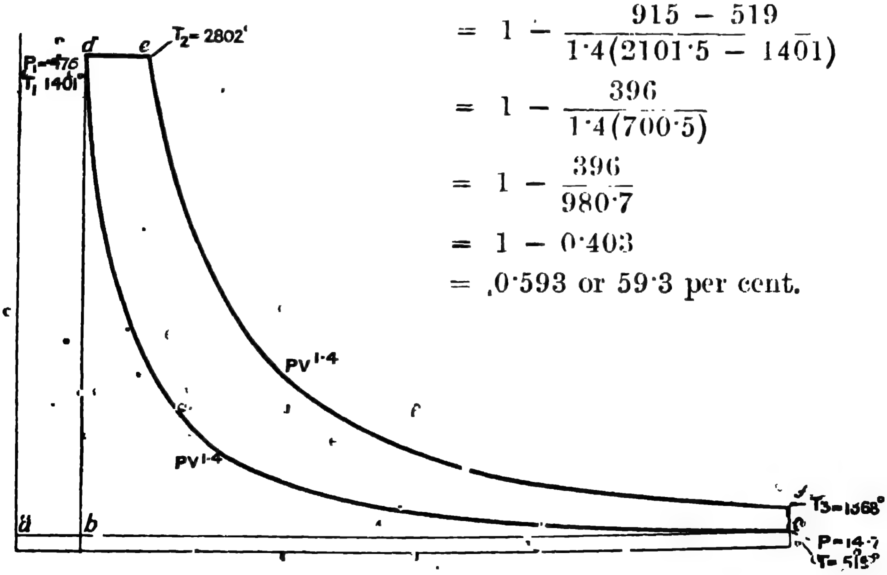


Fig. 5

Taking the second example (fig. 5), when the fuel is cut off at $\frac{1}{11}$th of the stroke. The volume at e becomes $\frac{1}{6}$th of the total, and the temperature T_2 at the point e is therefore

$$\frac{12}{6} \times 1401^\circ \text{ F.}$$
$$= 2802^\circ \text{ F. absolute.}$$

The temperature T_3 at the point f now becomes

$$\begin{aligned} T_3 &= T_2 \div 6^{\gamma-1} \\ &= 2802 \div 2{\cdot}048 \\ &= 1368^\circ \text{ F. absolute.} \end{aligned}$$

The efficiency in the second example is therefore

$$\begin{aligned} E &= 1 - \frac{K_v(T_3 - T)}{K_p(T_2 - T_1)} \\ &= 1 - \frac{1368 - 519}{1{\cdot}4(2802 - 1401)} \\ &= 1 - \frac{849}{1{\cdot}4(1401)} \\ &= 1 - 0{\cdot}433 \\ &= 0{\cdot}567, \text{ or } 56{\cdot}7 \text{ per cent.} \end{aligned}$$

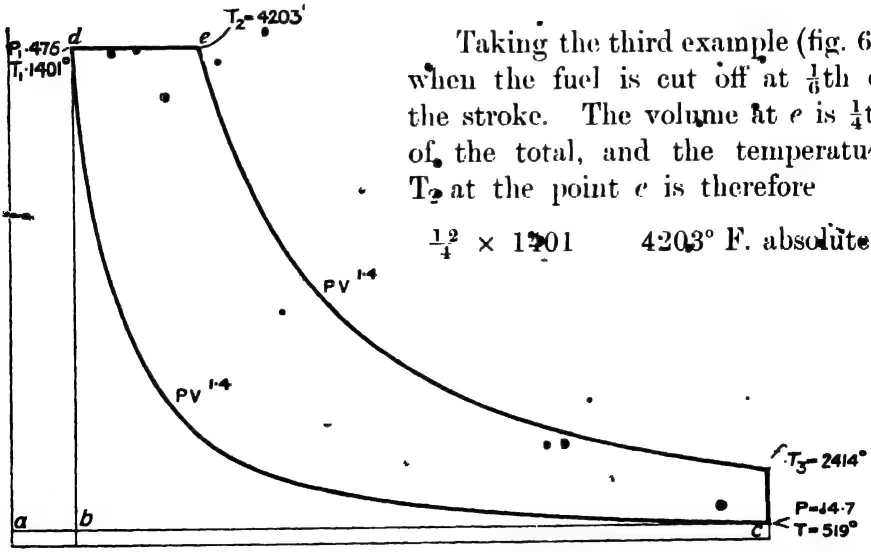


Fig. 6.

Taking the third example (fig. 6), when the fuel is cut off at $\frac{1}{6}$th of the stroke. The volume at e is $\frac{1}{4}$th of the total, and the temperature T_2 at the point e is therefore

$$\frac{12}{4} \times 1401 \quad 4203^\circ \text{ F. absolute.}$$

The temperature T_3 at the point f now becomes

$$\begin{aligned} T_3 &= T_2 \div 4^{\gamma-1} \\ &= 4203 \div 1{\cdot}741 \\ &= 2414^\circ \text{ F. absolute.} \end{aligned}$$

The efficiency in this case is therefore

$$\begin{aligned} E &= 1 - \frac{2414 - 519}{1{\cdot}4(4203 - 1401)} \\ &= 1 - \frac{1895}{1{\cdot}4(2802)} \\ &= 1 - 0{\cdot}483 \\ &= 0{\cdot}517, \text{ or } 51{\cdot}7 \text{ per cent.} \end{aligned}$$

If the same engine with the same compression ratio were operated on the constant-volume or explosion cycle the efficiency would be

$$\begin{aligned} E &= 1 - \left(\frac{1}{r}\right)^{\gamma-1} \\ &= 63 \text{ per cent,} \end{aligned}$$

but the maximum pressure in the latter case would be no less than

$$476 \times \frac{4203}{1401} \text{ lb. per square inch absolute,}$$

$$1428 \text{ lb. per square inch absolute.}$$

The Semi-Diesel Engine.—In addition to the Explosion type and Diesel type, there is, as already mentioned, an intermediate type which has recently become very popular, and is generally known as the Semi-Diesel. In this class of engine combustion takes place partly at constant pressure and partly at constant volume. Air only is compressed in the cylinder, but the compression is not carried to such a degree as in the true constant-pressure engine, and consequently it is not sufficiently heated by compression alone to ignite the fuel. In order further to heat the air and to ensure ignition of the fuel, a portion of the combustion chamber, generally in the form of a bulb, is left uncooled, and allowed to attain a high temperature. The fuel is sprayed into this bulb, and ignited partly by the heat of the air and partly by heat supplied to it by contact with the highly heated walls. Under these conditions, however, the combustion of the fuel is not sufficiently rapid, and it has been found desirable, in order to obtain complete combustion, to admit the fuel slightly before the end of the compression stroke. Since the fuel is present in the cylinder before the end of the compression stroke, it is clear that combustion of part at least takes place at constant volume.

Engines of this type cannot be fairly classified under either head, for they may be either constant-volume, constant-pressure, or any

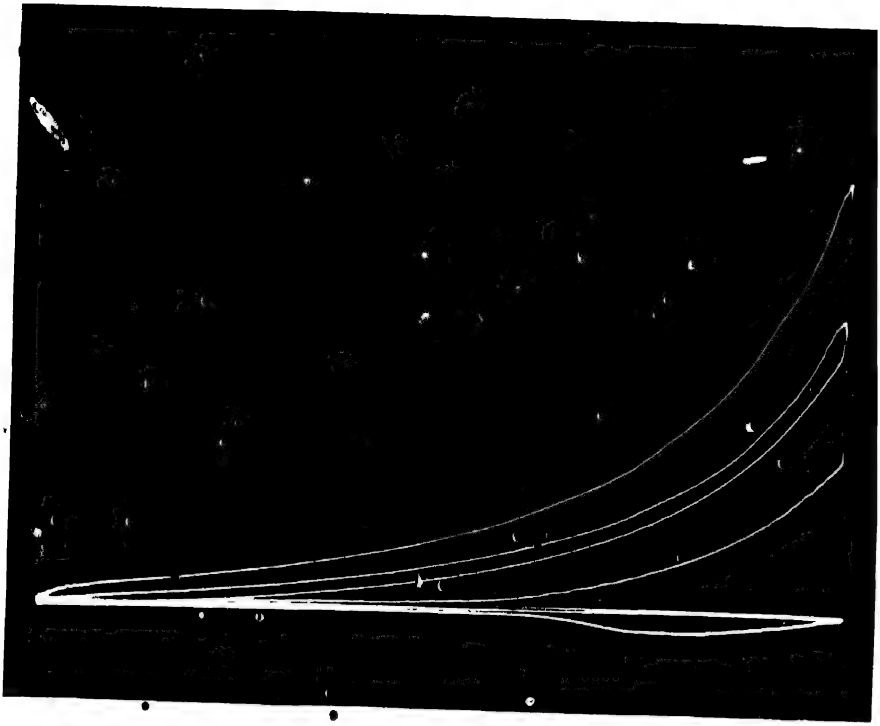
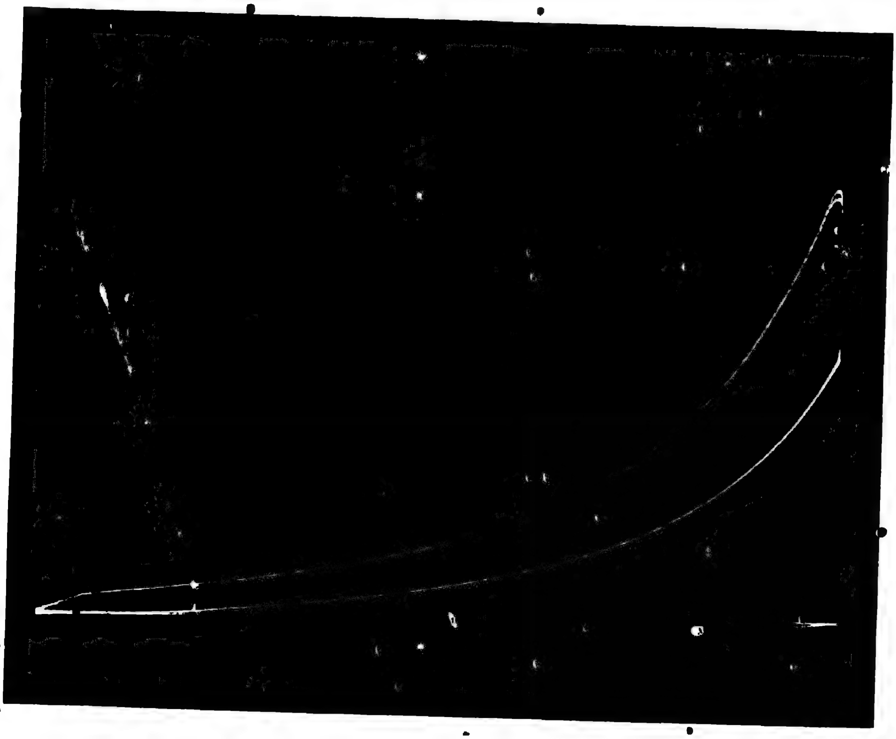


Fig. 8



compromise between the two, depending upon the exact period of the combustion of the fuel. In practice there is a very wide variation in this respect among the different types of Semi-Diesel engines, but, broadly speaking, they may be taken as about half-way between the two extreme types. It is clear that the nearer they approach to the explosion type the higher will be their efficiency, but the difference is not very great. This type of engine has many points in its favour, and bids fair to become extremely popular in the near future.

Comparison of Diesel and Explosion Types.—Examining the three different conditions of running of constant-pressure engines, it will be observed that the efficiency falls from 59·3 per cent when the maximum temperature is 2101·5° absolute to 51·7 per cent when the maximum temperature is 4203° absolute. It is improbable that a sufficient quantity of fuel could be burnt in the cylinder to produce a higher temperature than 4203° absolute. In practice the maximum temperature will not greatly exceed 3500° F., which would give a theoretical efficiency of about 54 per cent.

To sum up the conclusions so far arrived at. It is evident:—

1. That in an explosion engine the efficiency is entirely dependent upon the ratio of compression, and is independent of the temperature.

2. That in a constant-pressure or Diesel engine the efficiency is dependent mainly upon the compression ratio, but also to a limited extent upon the maximum temperature.

3. That for equal compression ratios the explosion engine is the more efficient type, but, for reasons that will be explained later, it is not possible to use such a high compression ratio with engines of this type.

Prolongation of Expansion Stroke.—In both types of engine which have been considered so far, expansion is carried only to the same volume as before compression; this is the case in almost every engine at present built, but attempts have been made to increase the expansion ratio in relation to the compression. Since there are indications that this may be achieved in some form in the near future, and indeed in the case of the Humphrey pump has already been achieved, it is interesting to calculate what the efficiency would be both in the case of the constant-volume and constant-pressure types, if expansion were continued down to atmospheric pressure.

Taking the constant-volume type first, in which the compression ratio equals 5:1, and referring to the indicator diagram, fig. 7, let

it be supposed that instead of the gases being released at *e*, the expansion is continued until their pressure has dropped to atmospheric at the point *f*. Then, if the pressures and temperatures throughout the rest of the cycle remain unchanged, the expansion line will meet the atmospheric line at the point *f*, and the number of expansions may be found from the equation

$$\begin{aligned} r &= \left(\frac{450}{14 \cdot 7}\right)^{\frac{1}{\gamma}} \\ &= 30 \cdot 61^{0 \cdot 71} \\ &= 11 \cdot 5 \text{ expansions.} \end{aligned}$$

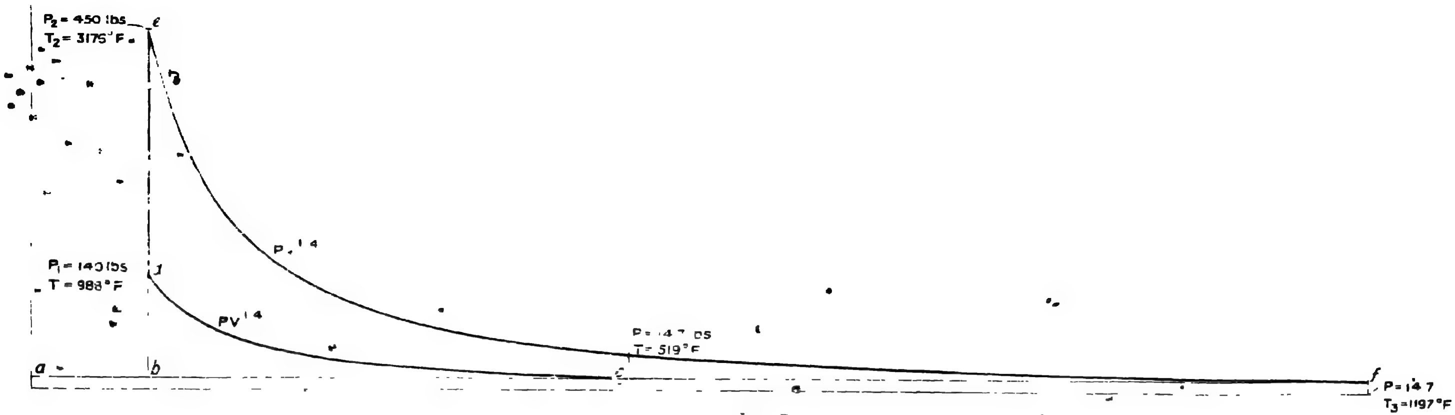
The length of the expansion stroke must therefore be in this case $\frac{11 \cdot 5}{5}$, or 2·3 times the length of the compression stroke. The temperature T_3 at the point F must now be found from the equation

$$\begin{aligned} T_3 &= T_2 \div 11 \cdot 5^{\gamma - 1} \\ &= 3176 \div 2 \cdot 656 \\ &= 1197^\circ \text{ F. absolute} \end{aligned}$$

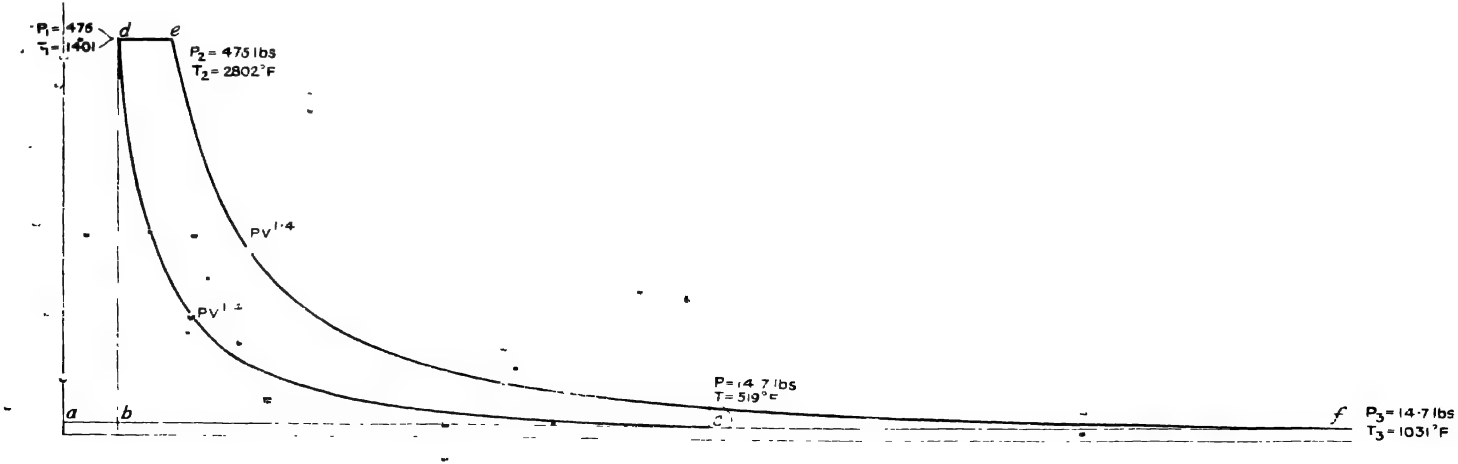
In this case the heat supplied to the cycle is $H = K_v(T_2 - T_1)$, the heat discharged is $H_1 = K_p(T_3 - T)$, and the efficiency is

$$\begin{aligned} E &= \frac{K_v(T_2 - T_1) - K_p(T_3 - T)}{K_v(T_2 - T_1)} \\ &= 1 - \gamma \frac{T_3 - T}{T_2 - T_1} \\ &= 1 - 1 \cdot 4 \, \frac{1197 - 519}{3176 - 988} \\ &= 1 - 1 \cdot 4 \, \frac{678}{2188} \\ &= 1 - 0 \cdot 433 \\ &= 0 \cdot 567, \text{ or } 56 \cdot 7 \text{ per cent.} \end{aligned}$$

That is to say, if the expansion of the gases be carried on until their pressure has dropped to atmospheric, the possible air standard efficiency will be increased from 48 per cent to 56·7 per cent, but it will no longer be dependent upon the compression ratio alone, and will vary according to the maximum temperature. The length of stroke required to expand the gases down to atmospheric pressure will also be dependent upon the maximum pressure and temperature, and to obtain the best possible efficiency, it will be necessary to vary the length of the expansion stroke in relation to the compression stroke, for every variation of the temperature and pressure,



P_2 = 450 lbs
T_2 = 3175°F
P_1 = 140 lbs
T = 988°F
PV 1.4
P = 14.7
T_3 = 1197°F
T = 519°F
a
b
d
e
f



P_1 = 476
T_1 = 1401
d
e
P_2 = 475 lbs
T_2 = 2802°F
PV 1.4
P = 14.7 lbs
T = 519°F
P_3 = 14.7 lbs
T_3 = 1031°F
a
b
f

which is impracticable in any ordinary type of reciprocating engine. The diagrams, fig. 8 (see Plate facing p. 22), are interesting in that they are actual diagrams taken from the author's experimental engine, in which the ratio of expansion could be varied while the engine was running. The upper diagram shows the expansion carried to the same volume as before compression, the ratio of compression being 4·75:1, and the theoretical efficiency approximately 47 per cent. The lower diagram has a compression ratio of 2·6:1, and the air standard efficiency in this particular case is also approximately 47 per cent. The efficiency actually measured was precisely the same in both cases, namely, 30 per cent.

Taking next the case of Diesel-type engines, and selecting the second example in which the fuel admission is continued during the first one-eleventh of the stroke. In order to expand the gases down to atmospheric pressure, it is clear that they will have to be expanded to twelve times the volume they occupied at the point *e* (fig. 9), which, in this case, is to double their original volume, and the temperature at the point T_3 can be found from the following equation:—

$$\begin{aligned} T_3 &= T_2 \div 12^{\gamma-1} \\ &= 2802 \div 2{\cdot}706 \\ &= 1031^\circ \text{ F. absolute.} \end{aligned}$$

Since both the compression and expansion are adiabatic and between the same limits of pressure, it follows that the efficiency, in this case, depends solely upon the degree of compression, and can be found from the equation

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1};$$

since r in this case is 12, the efficiency is

$$\begin{aligned} E &= 1 - \left(\frac{1}{12}\right)^{0{\cdot}4}, \\ E &= 1 - 0{\cdot}37 \\ &= 0{\cdot}63, \text{ or } 63 \text{ per cent.} \end{aligned}$$

The efficiency, therefore, has been raised from 56·5 per cent, when expanding down to the same volume as before compression, to 63 per cent when expanding down to atmospheric pressure. It is interesting to note that in this case the efficiency is independent of the maximum pressure or temperature, and is dependent

only upon the ratio of compression; that is to say, no matter how long the period of constant pressure be maintained, provided that the length of the expansion stroke is sufficient to expand the burning gases down to atmospheric pressure, the efficiency may be obtained from the formula

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}.$$

To sum up, therefore—1. The air standard efficiency of the explosion type, when the gases are expanded to the same volume as before compression, is dependent solely upon the ratio of compression.

2. The efficiency of the Diesel type, when the gases are expanded to the same volume as before compression, is dependent *mainly* upon the compression ratio, but also partly on the maximum temperature, the higher the latter the lower the efficiency.

3. The efficiency of the explosion type, when the expansion is carried to atmospheric pressure, is no longer dependent upon the ratio of compression alone, but is also dependent upon the maximum temperature and increases slightly with increase of temperature.

4. The efficiency of the Diesel type, when the gases are expanded to atmospheric pressure, is no longer dependent upon the maximum temperature and pressure, but solely upon the ratio of compression.

5. The efficiency of the Diesel type, when expanding down to atmospheric pressure, is precisely the same as that of the explosion type, when expanding to the same volume as before compression, provided that the compression ratios in both cases are the same.

These conclusions are all based on the following assumptions:—

1. That the working fluid is pure dry air.

2. That its specific heat remains constant at constant volume over the whole range of temperature employed.

3. That both the compression and expansion are adiabatic.

4. That combustion in the one case is instantaneous and takes place at constant volume, and in the second case, that it is so delayed as to produce a constant pressure.

5. That the gases are expanded to precisely the same volume as they originally occupied.

In actual practice none of these assumptions hold good, for:—

1. The working fluid is not pure dry air, but may also contain a proportion of other gases whose specific heats at constant volume and constant pressure do not bear the same relation as in the case of air.

2. It has recently been proved by Dr. Dugald Clerk and others that the specific heat of the working fluid is not constant over a wide range of temperature, but increases considerably at very high temperatures. This is a matter of the greatest importance, and has a very powerful influence upon efficiency. The curve illustrated in

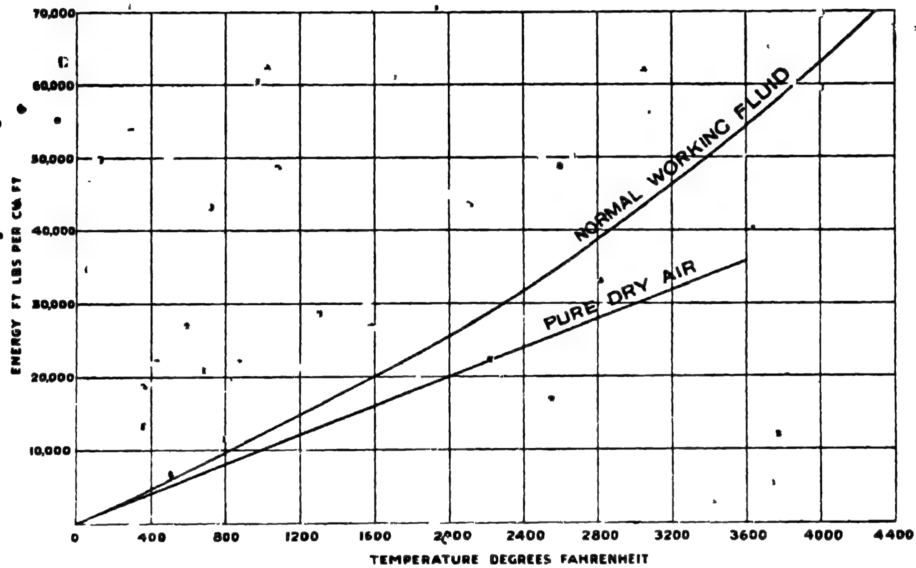


Fig. 10

fig. 10 shows the total amount of energy contained in the working fluid at various temperatures. The curve here shown is taken from that compiled by the Gaseous Explosions Committee of the British Association after a thorough investigation of the experimental results by Holborn and Henning, Clerk, Langen, Mallard, and Le Chatelier. More recent research suggests that the estimate of total internal energy at high temperatures is somewhat too high, and requires revision, but the values given are those generally accepted by engineers at present. It is based on the assumption that the analysis of the working fluid after combustion is—

N and O	83 per cent.
CO_2 ...	5 ,,
H_2O ...	12 ,,

3. Neither the compression nor expansion is truly adiabatic owing to the change that takes place in the specific heat of the working fluid. Also, heat is lost to the cylinder walls during both these strokes.

4. The combustion is not instantaneous, the propagation of flame throughout the whole of the working fluid takes an appreciable amount of time, and combustion continues during part, at least, of the expansion stroke.

5. Owing to the short time available for getting rid of the products of combustion it is necessary, in practice, to open the exhaust valve or ports slightly before the end of the stroke, and consequently the gases are not expanded to the same volume as they occupied before compression.

To make due allowances for all these discrepancies renders the calculation of the thermal efficiency of an engine exceedingly complicated, and necessitates an accurate knowledge of the chemical and physical characteristics of the working fluid.

Although, according to the air standard, the efficiency of explosion engines is entirely independent of the maximum temperature, when allowance is made for the increasing specific heat, it will be found that—other things being equal—the lower the maximum temperature the greater the efficiency; this is a matter of the greatest importance to the practical engineer, for the principal difficulties which he has to contend with are those due to the very high temperatures which obtain in the cylinders of internal-combustion engines.

CHAPTER II

PRINCIPAL SOURCES OF LOSS OF EFFICIENCY

Cooling Water Losses.—The possible efficiency of any type of engine being known, it now becomes necessary to investigate the various causes which prevent this efficiency being attained in practice. Besides the increase of specific heat at high temperatures, the loss of heat to the walls of the combustion chamber, cylinder, and piston account for a substantial proportion of the discrepancy. For obvious practical reasons, it is essential to keep these parts comparatively cool, and consequently a considerable proportion of the heat liberated during combustion and expansion passes into the cylinder walls, from which it is taken up by the cooling water, and is, to all intents and purposes, lost. In order to reduce these losses as far as possible it is essential that—

1. The surface exposed to the gases during combustion should be as small as possible, that is to say, the clearance or compression space should be as nearly spherical as possible.

2. The temperature of the working fluid should be as low as possible, consistent with a reasonably high mean pressure, for not only is the heat loss increased by the greater difference of temperature between the gases and the cylinder walls, but also, at very high temperatures, heat is lost to the walls by radiation also.

The actual proportion of heat lost to the cylinder walls, &c., during combustion and expansion is not easily determined, especially in four-cycle engines. If the amount of heat taken up by the cool ing water be measured, it will be found to give much too high a figure, because heat is being imparted to the cooling water, not only during combustion and expansion, but also during the exhaust stroke; this latter has, of course, no influence on the efficiency.

Dr. Dugald Clerk has, however, devised a most ingenious method for deducing the heat lost to the cylinder walls during the combustion and expansion alone. It is not proposed to describe this method here, but it may be stated broadly, that he has found the

true loss generally ranges between 40 and 60 per cent of that accounted for by the cooling water, the proportion, of course, depending upon the design, size, and speed of the engine, and many other factors. This proportion is very much lower than was popularly supposed to be the case some few years ago, and it is probable that it is not susceptible of much further reduction.

Slow Combustion.—Another cause of loss of efficiency, and one which is particularly serious in engines working on blast-furnace or other lean gases, is incomplete or delayed combustion. In the case of explosion engines, incomplete combustion is generally due to—

1. Stagnation of the working fluid at the time when combustion takes place.

2. The presence in the combustion chamber of pockets or recesses, which both obstruct the free flow of the gases at the time of combustion, and, by presenting a very large cooling surface in relation to the bulk of the gases which they contain, chill them, and so prevent or delay the propagation of flame.

3. The presence of a large proportion of inert gases, intermixed with the working fluid, such as exhaust gases, from the previous cycle, which tend to separate the particles of fuel and air.

In the case of constant-pressure engines using liquid fuel, incomplete or delayed combustion is generally due to want of thorough pulverization or distribution of the particles of fuel; with the result that each particle is not surrounded by a sufficient quantity of air for complete combustion.

Of all these causes, it is probable that stagnation of the working fluid is one of the most important. It is essential for complete combustion that the gases shall be in a state of rapid motion at the time when combustion takes place, in order that the flame shall be distributed mechanically throughout the whole mass of the working fluid; for the normal rate of propagation of flame is so slow that combustion will not be nearly completed, even by the end of the expansion stroke. Experiments on the combustion of stagnant gases in closed vessels have shown that the propagation of flame under such conditions is far too slow to be of any use in the cylinder of an internal-combustion engine. On the other hand, small petrol engines have been made to run quite satisfactorily and efficiently at speeds of over 4000 revolutions per minute, showing that if the velocity of the gases at the time of combustion is sufficiently high, the propagation of flame may be extraordinarily rapid.

The presence of a large proportion of inert gas delays the rate of flame propagation only when it is intimately mixed with the combustible mixture. This is a question which assumes very great importance in two-cycle engines, and will be considered in detail when dealing with this subject. If, by stratification, the combustible mixture can be prevented from mixing with the large bulk of inert gases, then their presence is desirable; for, let it be supposed that the quantity of working fluid in the cylinder is, say, 50 per cent of the cylinder volume, as it might be in the case of a four-cycle engine running partially throttled, and that the heating value is such that the maximum temperature obtained during combustion is, say, 3000° F.; if now, instead of throttling the gases, 50 per cent of inert gases be retained in the cylinder, but not mixed with the combustible gases, then, neglecting any difference in specific heat, the temperature on combustion will be only 1500° F., but the mean pressure and the work done will be the same. That is to say, double the quantity of working fluid will be heated to half the maximum temperature, but in this latter case, owing to the lower temperature, the losses will be considerably reduced, and the actual efficiency much greater. If, on the other hand, 50 per cent of inert gases be intimately mixed with an equal quantity of combustible mixture, it is probable that the resulting mixture will be so far diluted that combustion will not take place at all, or, if it does take place, will be so retarded as to be of little use.

In Diesel engines, as already stated, the completeness and rapidity of combustion is dependent upon the degree of pulverization and distribution of the liquid fuel throughout the highly compressed air in the combustion chamber. If the fuel is not sufficiently finely pulverized or distributed throughout the whole bulk of the air, then combustion is delayed until each particle of fuel can find the necessary quantity of oxygen. Pulverization is generally accomplished either by mechanical means or by compressed air; the latter is generally preferred, for not only are the particles of fuel more finely divided and distributed by compressed-air pulverization, but also the inrush of highly compressed air into the cylinder creates violent turbulence, and so increases the rapidity of the flame propagation. For this reason engines employing compressed-air pulverization generally show a somewhat higher efficiency, and are usually capable of running at a higher speed, than those in which the pulverization and distribution is effected purely by mechanical means.

Practical Limits of Compression.—It will be noted that although the explosion-type engine gives, for equal compression ratios, a higher efficiency, this advantage is not obtained in practice. It is not possible with this type to employ such a high compression ratio, because, if the combustible mixture be compressed beyond a certain point, the heat due to compression may ignite the charge before the end of the stroke, and thus throw very severe stresses upon the working parts of the engine. Since various gases have different ignition temperatures, it follows that, to obtain the best results the compression ratio must be varied to suit the particular fuel on which the engine is intended to run. As a general rule, it may be taken that the richer the gas the lower the ignition temperature, and therefore the lower the compression ratio that may be safely employed. There are, however, exceptions to this rule, such, for example, as alcohol vapour. Thus, for very rich gases, such as petrol vapour, a compression ratio of from 4 to 5 : 1 is generally used; for illuminating gas, or, as it is commonly called, town gas, the ratio is usually from 5 to 6 : 1; for producer gas from 6 to 7 : 1; and for blast-furnace gas compression ratios of from 7 to $7\frac{1}{2}$: 1 are sometimes employed, depending upon the percentage of hydrogen present in the gas. For very high percentages of hydrogen it is necessary to use a low compression ratio; thus, for coke-oven gas, which frequently contains over 50 per cent of hydrogen, it is hardly ever safe to use a compression ratio much higher than 5 : 1. In addition to the rise of temperature due to compression alone, there are other causes which are liable to produce premature ignition of the combustible charge, such for example as:

1. The presence of a large proportion of highly heated exhaust gases in the cylinder at the commencement of the compression stroke, causing a high initial or suction temperature.

2. The presence of uncooled parts of the engine, such as exhaust valves, &c., which are heated to a very high temperature during the combustion and expansion strokes, and impart their heat to the gases during compression.

3. The presence of a coating of carbon on the walls of the combustion chamber and piston head, which is a poor conductor of heat, and may attain a very high surface temperature during the expansion stroke, and impart much of the heat it has accumulated to the gases during compression, instead of conducting it through the cylinder walls to the water-jacket.

4. The presence of detached or partially detached portions of

carbon, igniter electrodes, and other parts, which are insulated from the water-jacket, and which in consequence may become incandescent.

5. In the case of certain fuels, and more particularly light volatile liquid fuels, such as paraffin or petrol, detonation occurs when the working fluid is compressed beyond a certain pressure. Such detonation is not in itself directly harmful, but there is evidence that if detonation be allowed to persist, it gives rise eventually to pre-ignition. There is evidence also that could means be found for preventing detonation, a much higher compression ratio could at once be used without risk of pre-ignition.

It is because of these causes that in practice a very much lower compression ratio must be employed than would at first sight appear necessary. The first cause applies in practice only to certain types of two-cycle engines, which on light loads retain a very large percentage of exhaust gases, and which in consequence are condemned to use a lower compression ratio than would otherwise be adopted.

It is obvious that for these and other reasons, which will be dealt with later, it is of the utmost importance that the contents of the cylinder at the commencement of the compression stroke should be at the lowest possible temperature.

In the constant-pressure type of engine, since air only is compressed in the cylinder, there is, under normal conditions, no risk whatever of pre-ignition, and consequently the only limit to the ratio of compression is the practical difficulty of employing very high pressures without considerable leakage and friction losses. In Diesel oil-engines a compression ratio of from 13 to 14 : 1 is now generally employed, but in certain special cases compression ratios as high as 18 : 1 have been successfully used.

Power and Cylinder Capacity.—Second only in importance to the efficiency is the power that can be obtained from a given size of cylinder, and this depends upon the weight of air that can be passed through the engine in a given time, and upon the efficiency with which it can be burnt. Whether the engine operates on the two- or four-stroke cycle, whether it be of the constant-volume or constant-pressure type, the ultimate power that it can develop depends primarily on the weight of air that it can deal with per minute, for it is evident that however much air is present in the cylinder, sufficient fuel can always be admitted to combine

with it, but that any further addition of fuel will be useless unless there is sufficient oxygen present for its combustion. In order that the engine shall be capable of dealing with the maximum weight of air in a given time the following conditions must be observed:

1. The piston speed must be as high as possible, consistent with a good mechanical efficiency.

2. The valves, together with their passages and ports, must be so designed as to offer the minimum of obstruction to the flow of gases.

3. The incoming charge must be at as low a temperature as possible in order that its density may be at a maximum.

That is to say, considered as an air-pump, the volumetric efficiency must be as high as possible. The conditions enumerated above are also the conditions required for high efficiency, but to obtain the highest possible mean pressures, and therefore the highest powers, very high temperatures must be employed, and these are not compatible with high thermal efficiency, for, as already explained, very high temperatures involve increased heat losses and an increase in the specific heat of the working fluid. From a purely thermodynamic point of view, high piston speeds are to be recommended, but their effect upon the mechanical efficiency and the wear and tear of the engine generally, will be dealt with later. As in the case of an air-compressor, the volumetric efficiency depends very greatly upon the valves; their size, position in the cylinder, and time of opening and closing are of the utmost importance, and, equally important, is the design of the passages leading to and from them. These should be free from sharp bends or sudden changes of area, and should if possible be designed so that advantage is taken of the inertia of the gases in the pipes and passages. It is not often necessary or desirable to use particularly large valves, but the pipe work and the contour of the passages immediately behind the valves should be so designed that the gases are led to the inlet valve at a gradually increasing velocity, which reaches a maximum at the opening of the valve itself. By this means advantage may be taken of the inertia of the gases in the inlet pipe to charge the cylinder fully, while their high velocity on entry maintains them in a state of violent turbulence during compression, thus ensuring rapid and complete combustion. The same remarks apply to the exhaust valve, but since in this case the gases are at a comparatively high pressure when the valve is first opened; their

velocity when released is exceedingly high, and still more advantage may be taken of their inertia thoroughly to empty the cylinder.

Condition 3, low temperature of the charge, can only be met in a four-cycle engine by reducing the proportion of residual exhaust gases to a minimum, for these gases, being at a very high temperature, impart their heat to the incoming charge, raising its temperature, and in consequence reducing its density. The proportion of residual exhaust gases can be reduced.—1. By paying particular attention to the exhaust outlet, and the timing of the exhaust valve in order to ensure that a perfectly free outlet is provided, and by taking advantage of the inertia of the gases in the exhaust pipe.

2. By employing some method of scavenging the combustion chamber, that is to say, of driving the exhaust gases out of the combustion chamber by means of a charge of air, under a light pressure, introduced before the commencement of the suction stroke.

Mechanical Efficiency.—Up to the present only the indicated efficiency and power have been considered, that is to say, the work done on the piston. To the practical engineer, the actual brake efficiency and power is of far greater importance, i.e. the efficiency and power developed at the crankshaft and available for external work. The ratio between the indicated power and the brake or available power is known as the mechanical efficiency, and this usually ranges in modern engines of the explosion type between 80 and 90 per cent, the remaining 10 to 20 per cent being absorbed in internal friction in the engine and in pumping or fluid losses. It is for the practical engineer to reduce these losses as far as possible. At the present time the indicated thermal efficiency of a modern internal-combustion engine, whether two- or four-cycle, explosion or Diesel type, is generally within from 10 to 20 per cent of what is theoretically possible, when due allowance has been made for the increasing specific heat at high temperature, and it does not seem that this is susceptible of much improvement, unless a new cycle be discovered, or unless some use be made of the heat lost to the cooling water or the exhaust.

To obtain the highest possible mechanical efficiency, the following conditions must be observed:—

1. The ratio of maximum to mean pressure must be as low as possible.

2. The piston and rotative speeds must be low.

3. The weight of the reciprocating parts must be reduced to a minimum.

4. The mean effective pressure must be as high as possible.

5. The moving parts of the engine must be, as nearly as possible, in perfect static and dynamic balance.

Influence of Maximum and Mean Pressure Ratios.—Taking Condition No. 1 first: It is of the utmost importance that the ratio of maximum to mean pressure should be as low as possible, because the working parts and the whole structure of the engine must be strong enough to withstand the maximum pressures, not only those that occur under normal running conditions, but also under abnormal conditions, such as premature ignition. If the ratio be high, then the weight, inertia, and the friction of the moving parts become excessive. This condition, as applied to explosion engines, is perfectly compatible with the thermodynamic one, that the maximum temperatures, and therefore pressures, should not be high, but it is not compatible with the demand for a high compression ratio, for, beyond a certain point, any increase in the compression ratio increases the maximum pressures enormously with very little increase in the mean pressure. In practice it has been found that very little improvement in the actual efficiency or power is obtained with increase of compression beyond a certain limit, because the maximum pressures become so high, and the working parts have to be so heavy in consequence, that any advantage is swallowed up by the increase of friction. Again, although, theoretically, a very material increase in the efficiency would be obtained if the gases were expanded down to atmospheric pressure, the ratio of maximum to mean pressure in that case would be so unfavourable that it is doubtful whether any appreciable advantage would be obtained in practice under normal circumstances.

Revolutions and Stroke.—Condition 2: The piston and rotative speeds must be as low as possible, for the greater part of the friction in an engine is due to the inertia of the reciprocating parts, which increases directly as the piston speed and as the square of the rotative speed. In order, however, to reduce the heat losses to a minimum, and also to obtain a reasonable power from a given size of cylinder, it is necessary to employ a high piston speed. These two conditions are conflicting, and a compromise must be sought. Since, as stated above, the friction increases directly as the piston speed, and as the square of the rotative speed, it is

obviously desirable to increase the piston speed without increasing the rotative speed; that is to say, the stroke of the engine should be as long as possible.

Weight of Reciprocating Parts.—Condition 3: The greater part of the mechanical friction is due to the inertia of the reciprocating parts. During the first half of the stroke work is done by the crankshaft in accelerating the piston, and during the second half this work is returned to the crankshaft and the balance restored. This interchange of work between the piston and crankshaft occurs twice every stroke, or in a four-cycle engine eight times every cycle, and each time a certain percentage is absorbed in friction. It is clear that for equal piston and rotative speeds the work done upon the piston and returned by the piston will depend upon the weight of that part, and the friction will be more or less proportional. In some modern engines running at high speeds, the friction due to the inertia of the reciprocating parts represents about 70 per cent of the total mechanical friction of the engine. It is obvious, therefore, that to obtain the highest possible mechanical efficiency the weight of the reciprocating parts must be reduced to a minimum.

Mean Effective Pressure.—Condition 4: The mean effective pressure must be as high as possible. This is fairly obvious; but to illustrate it, let it be supposed that a certain engine develops 50 indicated horse-power with a mean effective pressure of 50 lb. per square inch, and that the power absorbed in internal friction is 10 horse-power, then the brake or effective horse-power will be 40, and the mechanical efficiency $\frac{40}{50}$, or 80 per cent. If, now, the mean effective pressure be increased from 50 to 100 lb. per square inch, the engine will develop 100 indicated horse-power, and assuming that the internal friction is not materially altered, the B.H.P. now becomes $100 - 10 = 90$ horse-power, and the mechanical efficiency $\frac{90}{100}$, or 90 per cent.

Condition 5: This is a difficulty with single-cylinder engines. The importance of accurate balance and the conditions which govern its attainment will be dealt with later in this volume under the heading of Balancing. If the working parts are not in perfect balance, a portion of the power of the engine is dissipated in vibration.

Manufacturing Costs.—Of no less importance than the thermal and mechanical efficiency is the question of cost of manufacture. To be a practical success the cost of manufacture per horse-power must not be too high, and in many cases it is advisable to reduce the first cost even at the sacrifice of thermal efficiency. The

conditions necessary for low cost of production may be summarized as follows:—

1. The mean effective pressure must be as high as possible.
2. The maximum pressures must be as low as possible.
3. The stroke must be short.
4. The piston speed must be as high as possible.
5. The valve gear must be of the simplest type, and the valves so placed that they can be operated by the simplest form of gearing.
6. The engine should contain no intricate parts that cannot easily be made with ordinary tools, or which necessitate a degree of accuracy that is not obtainable in an ordinary engineering workshop.

Condition 1 is obvious, for the higher the mean pressure the greater the power that can be obtained from a given size of engine; this is also the condition required for good mechanical efficiency, but not for the best thermal efficiency, because a high mean pressure generally entails high temperatures.

Condition 2: The lower the maximum pressure the lighter and cheaper the engine; fortunately this is the condition required for good thermal and mechanical efficiency, in the case of explosion engines at all events.[1] In constant-pressure engines, where the degree of compression is not limited by the risk of pre-ignition, the higher the compression the higher the thermal efficiency, and the limit of compression is generally set by the first cost and mechanical efficiency. Apart from the fact that high pressures necessitate a very heavy and therefore costly engine, the risk of leakage is very great, and a much higher degree of accuracy is necessary in the manufacture, which, of course, increases the first cost.

Condition 3: The first cost of an engine depends very largely upon the stroke. An engine with a stroke equal to three times the diameter of the cylinder will cost nearly double as much as an engine in which the stroke is equal to one and a half times the diameter, for increase in stroke necessitates a corresponding increase in almost every other part of the engine, and the long-stroke engine will be nearly twice as bulky and heavy as the short-stroke. Therefore, from the point of view of first cost alone, the stroke must be kept as short as possible. This is not compatible with high thermal or mechanical efficiency, and therefore a compromise must be sought, depending upon whether low cost or high efficiency is most to be desired.

[1] So long as the maximum pressure is controlled by temperature conditions and not by the use of an abnormally low compression.

Condition 4: Since, other things being equal, the power of an engine varies directly as the piston speed, it is evident that any increase in the piston speed of a given engine will increase the power without increasing the cost. Unfortunately a high piston speed combined with a short stroke, and therefore a high rotative speed, is the very worst condition from the point of view of mechanical efficiency; here again the conditions favouring low cost of production and mechanical efficiency are conflicting, and it is necessary to compromise.

Condition 5: From a purely thermodynamic point of view, to obtain the best possible efficiency, the interior of the combustion chamber should be hemispherical, and there should be no recesses for the valves; in practice this means that the valves must be inclined at an angle to one another in the combustion chamber; such an arrangement is very costly, not only as regards the fitting of the valves themselves, but also the valve gearing. It is occasionally employed for engines in which first cost is not a serious consideration, such as aeroplane and racing automobile engines, but as a general rule, it may be dismissed as being altogether too costly and cumbersome for ordinary commercial use. There are numerous arrangements of valves and valve gearing which are simple and inexpensive, and which fulfil the required conditions sufficiently nearly for all practical purposes.

Condition 6: It is important that the engine should be free from intricate parts, which are troublesome to make or require special tools, nor should the engine require a higher standard of workmanship than is obtainable in any ordinary engineering workshop. Every year, however, the general standard of workmanship and accuracy shows a steady improvement, so that the importance of this condition is diminishing, especially if the engines be made in large quantities, and the parts can be thoroughly standardized.

CHAPTER III

ANALYSIS OF A MODERN GAS ENGINE

Investigation of Losses under Working Conditions.—Having set down some of the conditions necessary for the best thermal and mechanical efficiency, and also for low cost of production, it is interesting to take a typical example of a modern commercial engine, and examine it in detail to see how these conditions are complied with, and, when conflicting, what compromises should be made. The engine illustrated in section in fig. 11, may be regarded as a typical example of a first-class modern gas engine of the explosion type, operating on the four-stroke cycle, and designed with a view to obtaining a high efficiency with due regard to manufacturing cost. Let it be supposed that the engine has the following leading dimensions:—

1. Bore	12 in.
2. Stroke	18 in.
3. Rotative speed	240 R.P.M.
4. Piston speed	720 ft. per minute.
5. Compression ratio (r)	6 : 1.
6. Area of piston	113·1 sq. in.
7. Ratio of swept volume to clearance volume	5 : 1
8. Weight of reciprocating parts	600 lb.
9. Weight of reciprocating parts per square inch of piston	5·3 lb.

Taking first the indicated efficiency and power; since the compression ratio (r) is 6 : 1, it follows that the theoretical indicated thermal efficiency according to the air standard is

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

$$= 1 - \left(\frac{1}{6}\right)^{\gamma-1}$$

$$= 1 - 0{\cdot}489$$

$$= 0{\cdot}511, \text{ or } 51{\cdot}1 \text{ per cent.}$$

This is the highest possible efficiency, assuming that combustion is perfect and instantaneous, that there is no loss of heat to the cylinder walls, and that the working fluid is pure dry air, whose specific heat is constant over the whole range of temperature. In practice, such an engine will probably show an indicated thermal efficiency of about 36 per cent, under the most favourable conditions, or 70·4 per cent of the air standard efficiency, which is a result that a good modern four-cycle engine might be expected to obtain.[1]

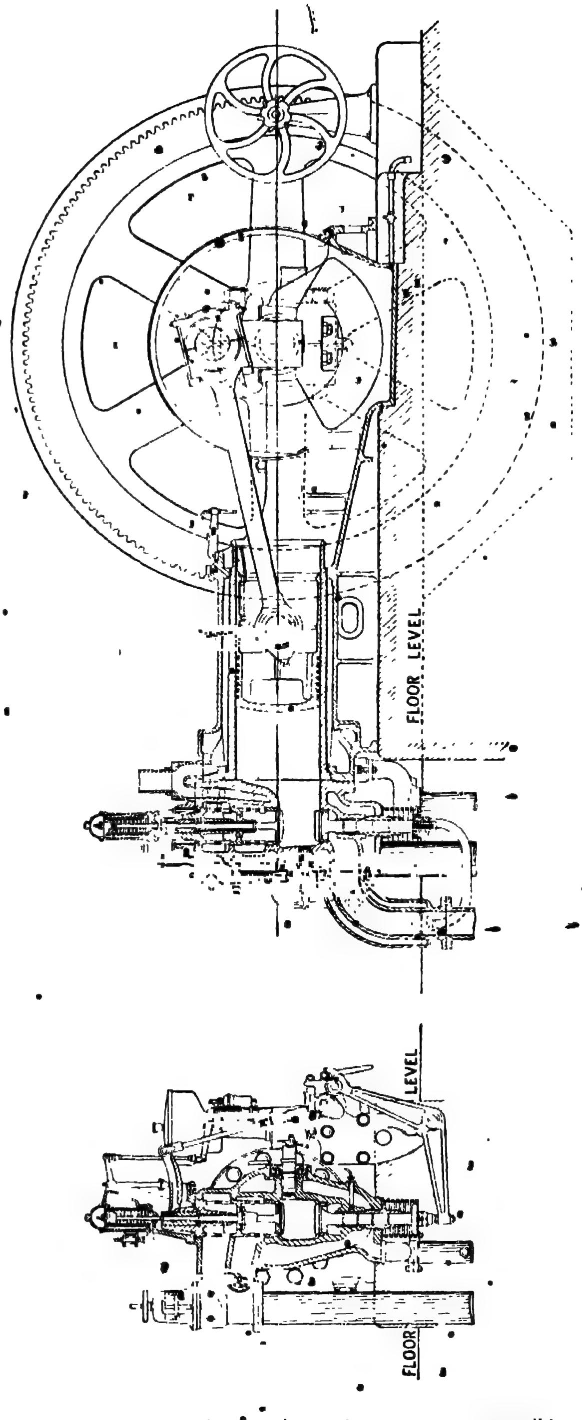


Fig. 11.—Sectional Arrangement - Horizontal Gas Engine (Crossley Bros., Ltd.)

It is interesting to endeavour to trace the 29·6 per cent discrepancy between the actual and ideal efficiencies. To do this, it is convenient to construct an indicator diagram (fig. 12), and to find the temperatures at the points c, d, e, and f. It is a difficult matter to determine

[1] When working with best illuminating gas, and using the weakest mixture compatible with complete combustion.

the exact temperature at the point c, that is to say, the temperature of the working fluid within the cylinder at the moment when the suction stroke is completed, and the compression stroke about to commence. It must be remembered that, not only does the incoming charge take up a small amount of heat from the walls of the cylinder and the valves, but it is also mixed with the exhaust gases retained in the clearance space. Professor Hopkinson, in his paper on the "Thermal Efficiency of Gas Engines", read before the Institution of Mechanical Engineers, arrives at the following conclusions with regard to the suction temperature. He calculates that the temperature of the exhaust gases retained in the cylinder is approximately 1080° F., or 1550° F. absolute, and their bulk at a temperature of 32° F. will therefore be

$$\frac{491}{1550} \times 0{\cdot}2 = 0{\cdot}063 \text{ of the swept volume.}$$

The quantity of fresh charge taken in during the suction stroke was found, in the particular engine he was testing, to amount to 80·5 per cent of the swept volume at normal barometer and a temperature of 32° F. The total contents of the cylinder therefore consist of 6·3 per cent of exhaust gases and 80·5 per cent of fresh charge, a total of 86·8 per cent of the swept volume. The total volume of the cylinder, including the clearance or compression space, is 120 per cent of the swept volume, and this is filled with air and gases which, at atmospheric pressure and at a temperature of 32° F., or 491° absolute, occupy only 86·3 per cent of the swept volume; their temperature must therefore be

$$\frac{120}{86{\cdot}3} \times 491,$$

$$= 679^\circ \text{ absolute, or } 220^\circ \text{ F.}$$

This result is in very fair agreement with the direct thermometric measurements carried out by Professors Callendar, Dalby, Coker, and others. It is evident that the lower the compression ratio the greater the quantity of exhaust gases retained in the cylinder, and that, therefore, the temperature of the mixed gases within the cylinder at the point c will be higher. In this particular instance, if the temperature of the contents be assumed to be 220° F., the error, if any, will not be large. It must, of course, be understood that this applies to full-load running conditions only. If the engine be throttle-governed, the quantity of charge drawn into the

cylinder on light loads will be much reduced and the temperature will therefore be higher. On the other hand, if the engine be governed by missing explosions, then the charge drawn in on light loads, following a missfire, will be at a lower temperature, since it will be mixed with air only and no products of combustion.

Taking the temperature at the point c as 220° F., or 679° absolute, it is now necessary to find the temperature at the point d. From c to d the compression is not truly adiabatic, because during the latter part of this stroke heat is given up to the cylinder walls. Experience has shown that with engines of about this size a tolerably correct result will be obtained if the value of the index be taken as 1·35. In this case the temperature at the point d can be found from the formula

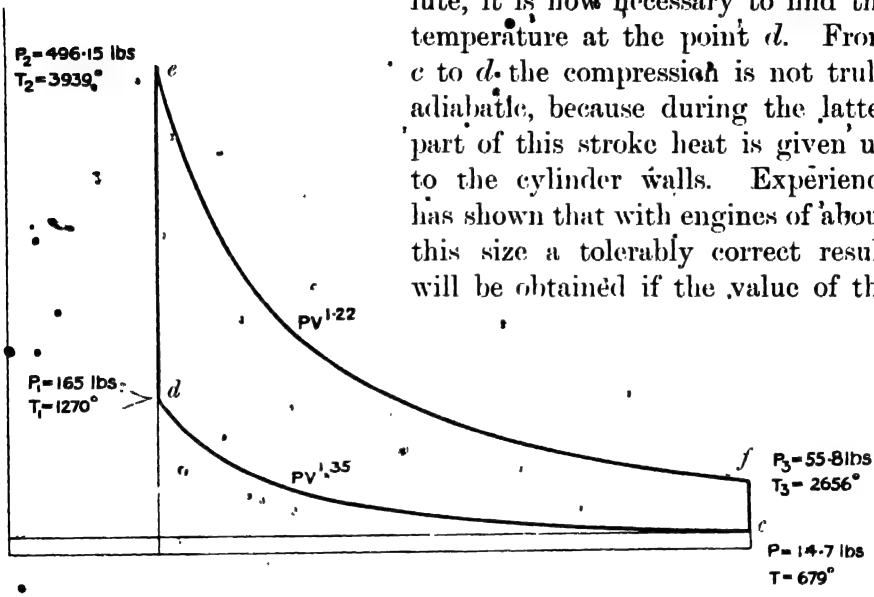


Fig. 12

$$T_1 = T \times 6^{\gamma-1}$$
$$= 679° \times 6^{0·35}$$
$$= 1270° \text{ F. absolute,}$$

and the pressure from the formula

$$P_1 = P \times 6^{\gamma}$$
$$= 14·7 \times 6^{1·35}$$
$$= 14·7 \times 11·23$$
$$= 165 \text{ lb. per square inch absolute.}$$

The rise of temperature during compression amounts to 591° F., and the work done on the gases can be calculated from the curve illustrated in fig. 10, which gives the internal energy per cubic foot of the working fluid at varying temperatures, and which takes into

account the increase of specific heat with increase of temperature. For the sake of simplicity, let it be supposed that the cylinder contains 1 cu. ft. of gases, of which approximately 92·7 per cent by volume is combustible mixture, and 7·3 per cent residual exhaust gases. Starting at 679° absolute, this mixture has been compressed to 1270°, or through a rise of temperature of 591° F. The thermal capacity, calculated from the specific heat of the gases given by the internal-energy curve, may be taken as 10·5 foot-pounds per cubic foot per degree Fahrenheit within the range of temperature of the compression stroke. The total work done on the gases during compression, therefore, amounts to

$$10{\cdot}5 \times 591 \text{ foot-pounds} = 6205 \text{ foot-pounds.}$$

To sum up. Between the points c and d the gases have been compressed into one-sixth of their original volume, their pressure has been raised from 14·7 lb. per square inch absolute to 165 lb. per square inch, their temperature from 679° absolute to 1270°, and the work done upon them amounts to 6205 foot-pounds.

Let it be assumed that the fuel used is ordinary illuminating or town gas, having a calorific value of 600 British Thermal Units per cubic foot, and that the proportion of air to gas is as 9 : 1; that is to say, 10 per cent of the mixture is gas and 90 per cent air. To this mixture there has been added 7·3 per cent of inert gases, so that the actual proportion of gas in the cylinder is

$$10 \times 92{\cdot}7 = 9{\cdot}27 \text{ per cent.}$$

The actual volume of gas, therefore, amounts to 0·0927 cu. ft. The heating value of the gas amounts to

$$0{\cdot}0927 \times 600 = 55{\cdot}6 \text{ B.T.U.s, or } 43272 \text{ foot-pounds.}$$

To this must be added 6205 foot-pounds, the work of compression making the total heat available at the point B

$$\begin{array}{r} 43272 \\ 6205 \\ \hline 49477 \end{array} \text{ foot-pounds.}$$

After ignition the volume of the products of combustion is slightly reduced, the contraction being about 3 per cent when reduced to the same temperature and pressure as before; that is to

say, the 1 cu. ft. will after combustion be reduced to 0·97 cu. ft., and its internal energy will be

$$49477 \times \frac{1}{0 \cdot 97} \text{ foot-pounds per cubic foot}$$
$$= 51007 \text{ foot-pounds per cubic foot.}$$

From the curve of internal energy of the working fluid (fig. 10) it will be seen that the corresponding temperature is 3480° F., or 3939° F. absolute. The pressure at the point e is

$$0 \cdot 97 \times \frac{3939}{1270} \times 165 \text{ lb. per square inch absolute}$$
$$= 496 \cdot 15 \text{ lb. per square inch absolute.}$$

Thus the maximum pressure is 496·15 lb. per square inch absolute and the maximum temperature is 3939° F. absolute. From e to f the highly heated gases are expanded, doing work on the piston. The expansion curve will follow the law, $pv^{\gamma} = c$, but, on account of the nature of the working fluid, and the practical conditions, the value of γ for adiabatic expansion will not be equal to 1·4. The true value of the index depends, of course, upon the specific heat of the gas, which varies according to the temperature; consequently a mean value must be found, which will be approximately correct between the limits of temperature during the expansion. Such a value can only be found by trial and error, and for this particular mixture if the value be taken as 1·22, it will give a very close approximation to the true expansion curve. The temperature T_3 at the end of the expansion will therefore be

$$\begin{aligned} T_3 &= T_2 \div 6^{\gamma-1} \\ &= 3939 \div 6^{0 \cdot 22} \\ &= 3939 \div 1 \cdot 483 \\ &= 2656^{\circ} \text{ F. absolute, or } 2197^{\circ} \text{ F.} \end{aligned}$$

The pressure at the point f will be

$$\begin{aligned} P_f &= P_2 \div 6^{\gamma} \\ &= 496 \cdot 15 \div 6^{1 \cdot 22} \\ &= 496 \cdot 13 \div 8 \cdot 895 \\ &= 55 \cdot 8 \text{ lb. per square inch absolute.} \end{aligned}$$

Referring again to the curve of internal energy, it will be seen that the total energy of 1 cu. ft. of the working fluid at a tempera-

ture of 2197° F. is approximately 26500 foot-pounds per cubic foot. Allowing for contraction, the total volume of the gases in the cylinder after combustion is 0·97 cu. ft., therefore the internal energy of the working fluid at the end of the expansion stroke is

$$26500 \times 0{\cdot}97 \text{ foot-pound} = 25905 \text{ foot-pounds.}$$

The total internal energy of the working fluid at the point e was 49477 ft.-lb., so that the energy exerted on the piston during the expansion stroke is

$$49477 - 25705 = 23772 \text{ foot-pounds.}$$

Of this, however, 6205 foot-pounds were absorbed in compressing the gases before combustion, so that the net useful work amounts to

$$23772 - 6205 = 17567 \text{ foot-pounds.}$$

Now the total energy obtainable from the heating value of the gas was 43272 foot-pounds, and the thermal efficiency is therefore

$$\frac{17567}{43272} = 40{\cdot}6 \text{ per cent.}$$

This then is the ideal efficiency which might be expected from an engine having a compression ratio of 6 : 1, and working with a 10-per-cent mixture of gas and air.

If the volume of gases in the cylinder at the end of the suction stroke, when reduced to standard pressure and temperature, was exactly 1 cu. ft., then the actual useful energy available would be 17567 foot-pounds. In the particular engine now under consideration, the actual volume of the gases retained in the cylinder at the end of the suction stroke will be slightly less than one standard cubic foot, for reasons connected with the volumetric efficiency which will be explained later. In an engine of these dimensions the volume retained in the cylinder may be taken as 0·97 standard cubic foot, and the available energy will therefore be approximately 17050 foot-pounds.

To obtain the mean pressure on the piston, all that is needed is to divide the number of foot-pounds usefully employed during the expansion stroke by the stroke of the piston in feet:—

$$\frac{17050}{1{\cdot}5} = 11366 \text{ lb.}$$

Expressed in pounds per square inch the mean effective pressure is

$$\frac{11366}{113 \cdot 1} = 100 \cdot 5 \text{ lb. per square inch.}$$

The indicated horse-power can now be arrived at as follows:—The engine being four-cycle and single-acting, there is one power stroke in every four, that is, in every two revolutions. At 240 R.P.M. there are 120 power strokes per minute, and the effective energy of each stroke is 17050 foot-pounds, so that the indicated horse-power will be

$$\frac{17050 \times 120}{33000} = 62 \cdot 0 \text{ indicated horse-power.}$$

Heat Losses—Influence of Strength of Mixture.—To sum up these results. It is clear that if combustion were instantaneous and complete, and if no heat were lost to the cylinder walls during combustion or expansion, then with a 10-per-cent mixture of gas and air this engine should develop 62·0 indicated horse-power. Its indicated thermal efficiency should be 40·6 per cent, and its mean effective pressure should be 100·5 lb. per square inch.

With the engine in question, which may be regarded as typical of the best modern practice, an indicated thermal efficiency of about 35 per cent or $\frac{35}{40 \cdot 6} = 86$ per cent of the possible efficiency may be expected; that is to say, of the 17050 foot-pounds available, about 14700 will be converted into useful work, and about 2350 will be lost. Better results than this have been recorded, but generally under rather exceptional conditions. Of the 2350 foot-pounds, or 14 per cent, unaccounted for it is probable that about 12 per cent is lost as heat to the cylinder walls, the remaining 2 per cent is to be accounted for by incomplete combustion, and the partial opening of the exhaust valve before the end of the expansion stroke. If, at the same time, the amount of heat carried away by the cooling water were measured, it would be found to amount to from 25 per cent to 28 per cent of the total heating value of the fuel, and this has led engineers to believe that the heat loss to the cylinder walls during combustion and expansion is very much greater than it really is.

Recent researches by Dr. Dugald Clerk, Professor Hopkinson, and others, have proved beyond all possibility of doubt that the heat loss during combustion and expansion usually does not exceed

about 12 per cent in the case of an engine of this size, using a working fluid of this density. The remaining 13 per cent to 16 per cent is imparted to the cylinder walls during the exhaust stroke; more particularly to the walls immediately surrounding the exhaust valve, and that part of the exhaust pipe which is included in the combustion head casting, and therefore water-jacketed. At these points the velocity of the exhaust gases is exceedingly high, and they consequently yield up their heat very much more rapidly, but this heat has already been accounted for and included in the exhaust losses. There is also the slight loss of heat to the cylinder walls during the compression stroke, but the amount is so small that it is hardly worth considering. Tests made on two-cycle gas engines, in which the exhaust takes place through ports and is generally cooled from a separate source of supply, show that the loss of heat to the cylinder walls is generally from 14 per cent to 16 per cent, which is in tolerable agreement with Dr. Clerk and Professor Hopkinson's conclusions.

The actual results that would be obtained from this engine are as follows:—

Indicated horse-power	... ...	53·5.
„ thermal efficiency	...	·35 per cent.
„ mean pressure	... ...	86·4 lb. per square inch.

If, instead of a 10-per-cent mixture, a 13-per-cent mixture were employed, the ideal efficiency would fall from 40·6 per cent to about 36·5 per cent, and the actual efficiency would fall to about 30 per cent. The drop in the ideal efficiency is due to the greater specific heat of the gases at the higher temperature, and the still greater drop in the actual efficiency to the larger proportion of heat lost to the cylinder walls, due also to the higher temperature of the gases. In this case, without going through all the necessary steps of the calculation, the heat supply will be 54500 foot-pounds, plus 6032 foot-pounds for compression, making the total energy of the gases 60532 foot-pounds, or 67500 foot-pounds per cubic foot, allowing for contraction after combustion. From the curve of internal energy, the corresponding temperature will be 4650° F. absolute, and the maximum pressure will be 604 lb. per square inch absolute. The temperature at the end of expansion will be 3390° F. absolute, and the energy lost to exhaust will be 34600 foot-pounds. Of this, 19900 foot-pounds will be usefully employed, thus giving a theoretical efficiency of 36·5 per cent. In this case,

however, the higher temperatures ruling in the cylinder during the expansion stroke will increase the heat loss to the cylinder walls, and it is probable that the actual efficiency will not be more than 85 per cent of the ideal efficiency.

The *actual* results that may be expected with a 13-per-cent gas-and-air mixture are:—

Indicated horse-power	... ...	61·5
" thermal efficiency	...	31 per cent.
" mean effective pressure	...	99·5

If a much richer mixture than 13 per cent be employed, it is

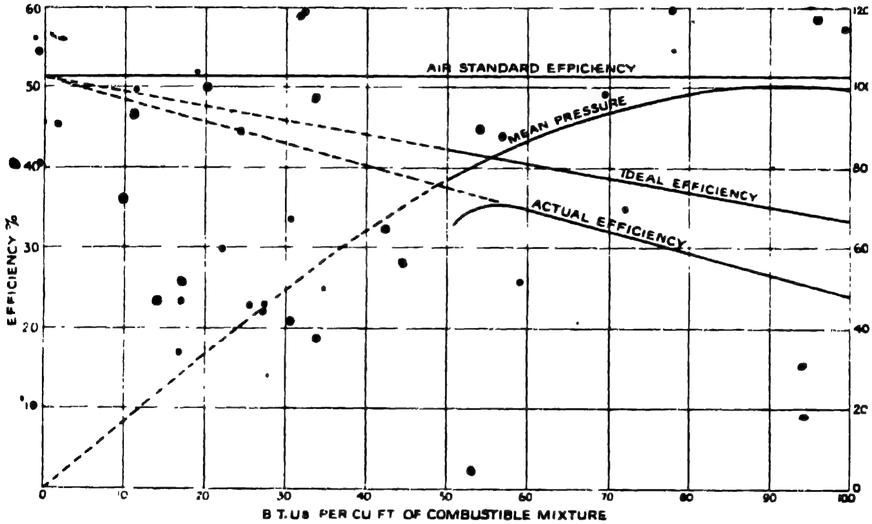


Fig. 13.—Volumetric Efficiency taken as 75 per cent in all cases when expressed in terms of standard pressure and temperature

probable that there will not be sufficient oxygen present for complete combustion, and the efficiency will fall away rapidly with any further increase in the proportion of gas.

These figures serve to emphasize the fact that, although the air standard efficiency is the same for all mixture strengths, the true ideal efficiency depends very greatly upon the proportion of gas present, and falls away rapidly when rich mixtures involving very high temperatures are employed. They also prove that the highest indicated efficiency will be obtained with the weakest possible mixtures and the lowest maximum temperatures. Unfortunately, however, if much weaker than 9 per cent, the mixture may fail to ignite, and combustion will either not take place at all, or be so seriously delayed as to be incomplete, even at the end of the

expansion stroke. The curves illustrated in fig. 13 give (*a*) the air standard efficiency for this engine, which, since it takes no account of the changes in the specific heat of the working fluid, remains constant throughout the full range of mixture strength; (*b*) the ideal efficiency as calculated above from the curve of internal energy, taking account of changes in specific heat; this latter is approximately a straight line, and meets the air standard efficiency line at the point of no-heat supply, that is when the working fluid is pure air; (*c*) the actual indicated thermal efficiency as obtained from a modern high-class engine of this size; and (*d*) the actual mean effective pressure.

Although the actual and ideal thermal efficiency curves rise steadily as the strength of mixture is reduced, the range of mixture strength over which it is possible to obtain rapid and complete combustion is very limited, and is indicated approximately for town gas by the full lines drawn on the actual efficiency curve. From these it will be noted that the highest ideal efficiency obtainable with this engine is about 41·5 per cent. The percentages of mixture strength along the horizontal line of the curve are given in terms of B.T.U.s per cubic foot. The efficiency curves are calculated for average town gas, but they will be found to be approximately correct for fuels of any heating value within certain limits. If it were possible by some means to burn weaker mixtures, it is clear that a higher efficiency would be obtained. Suppose, for the sake of illustration, the cylinder to be filled with pure air, or inert gases, and that a paper bag containing a 10-per-cent mixture of gas and air were inserted. Then, if at the end of the compression stroke the mixture were ignited, and the bag burst, liberating the burning gases in an excess of pure air, or even products of combustion, the ideal efficiency would be very much higher. If the contents of the paper bag amounted to 20 per cent of the cylinder volume, then the mixture after combustion, and the bursting of the bag, would be $\frac{20}{100} \times \frac{10}{100} = 2$ per cent gas and 98 per cent air. Under these conditions the ideal efficiency would be nearly 49 per cent, and since the maximum temperature would only be 1740° F. absolute, or 1280° F., the loss of heat to the cylinder walls would be exceedingly small, and an actual thermal efficiency of between 45 per cent and 46 per cent might be obtained. It must be noted, however, that under these circumstances the mean effective pressure would only be about 19 lb. per square inch,

and the indicator card obtained would be somewhat as shown in fig. 14.

A somewhat similar result might be achieved by means of stratification; that is to say, by so charging the cylinder that, while the main body of the combustion space contains only inert gases, a small proportion of readily combustible mixture is retained in the neighbourhood of the igniter. Recent experience has gone far to show that stratification, within certain limits, is possible if the combustion space be correctly designed, and the author confidently believes that higher efficiencies will be obtained in the near future by the employment of this principle. The indicator diagram, fig. 15

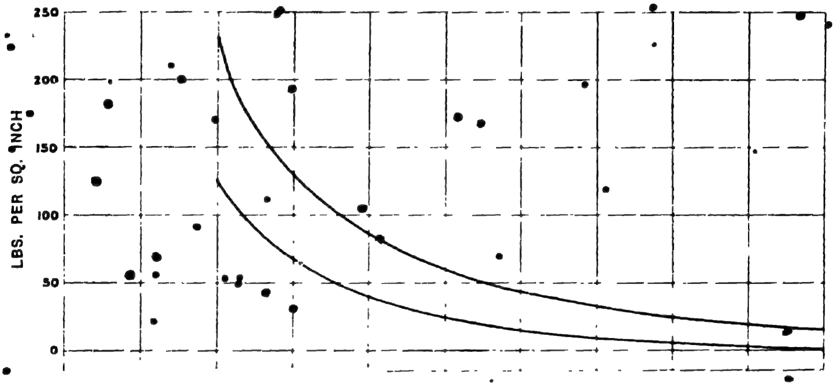


Fig. 14

ig. 15. See Plate facing page 22.

actual diagram obtained from the author's experimental engine when working under these very conditions.

All the preceding results have been calculated on the assumption that the temperature of the charge at the commencement of the compression stroke is 220° F., and that its absolute pressure is 14·7 lb. per square inch; that is to say, that the cylinder is completely filled with a mixture of fresh charge and products of combustion at the end of the suction stroke. If a smaller charge were taken in, then, to obtain the same mean pressure and indicated horse-power, a richer mixture and higher temperatures would have to be employed, with consequent loss of efficiency. It is therefore of the utmost importance that the volumetric efficiency of the cylinder shall be as high as possible. It is also equally important that the incoming charge shall be at a low temperature, for the lower the temperature the greater the density and weight of charge that can

be taken in. Also, if the temperature at *c* be lower, the whole range of temperatures throughout the cycle will be correspondingly reduced, or, alternatively, a higher compression may be safely employed without risk of pre-ignition or increase in the temperatures. In four-cycle engines using a mixture of gas and air the temperature cannot well be lower than that of the surrounding atmosphere, which is generally taken as 60° F. But when petrol is used as fuel, the rapid evaporation of the finely atomized particles of fuel both in the carburettor and in the cylinder has the effect of lowering the initial temperature and so increasing the weight of charge that can be introduced. Consequently a higher mean pressure can be obtained without any increase of temperature, and this, in part, accounts for the remarkably high mean pressures and efficiencies which are obtained from modern petrol engines. With two-cycle engines, when the working fluid is forced into the cylinder by means of a pump, and therefore at a somewhat high temperature, very beneficial results have been obtained by the introduction of an inter-cooler between the pump and power cylinders.

From all the above considerations it is evident that the thermal efficiency of a modern four-cycle explosion engine is not susceptible of any great improvement, at all events when running under full load. The heat loss to the cylinder walls, amounting as it does when running on nearly full load, and under favourable conditions as to mixture strength, to only about 12 per cent of the total heat of the fuel, is not susceptible of any great reduction. This loss must depend upon the difference of temperature between the gases and the cylinder walls, and under no conditions can the temperature of the latter be increased beyond a comparatively low figure without imperilling the efficient lubrication of the piston. It is probable that the greater part of this heat loss is imparted to the walls of the combustion chamber, which are exposed to the gases at the time of maximum pressure, and it is evident that in order to reduce this loss to a minimum the ratio of surface to volume should be kept as small as possible. This is particularly important in small engines, in which the ratio of surface to volume is necessarily great. The difference in exposed surface to volume, however, between the most efficient and the most convenient form of combustion chamber is not very great, and since the loss of heat even to this part does not represent a large percentage, the design of the combustion chamber is generally dictated by other considerations, which will be dealt with later.

To obtain further increases in the thermal efficiency of this engine means must be found for:

1. Increasing the weight of air that can be taken into the cylinder per cycle, either by cooling, scavenging out the residual gases, or supercharging the cylinder, or, of course, any combination of these three methods.

2. By making the utmost use of stratification, which, combined with supercharging, would enable high mean pressures to be obtained with low maximum temperatures and pressures.

3. By utilizing the heat rejected to the exhaust either for the generation of steam, or by the use of some form of regenerator applied to the working fluid itself.

4. By increasing the speed of rotation, so that the time during which the highly heated gases are in contact with the walls of the cylinder per stroke is reduced. The reduction of heat loss, however, will not in this case be anything like proportional to the increase of speed, because with higher speeds there is greater turbulence in the working fluid, and in consequence it will impart its heat to the cylinder walls at a greater rate. There are, however, other reasons in favour of an increase of piston and rotative speeds. The length of stroke also has a certain bearing upon the thermal efficiency, because a longer stroke generally permits of higher piston speeds, and a more compact clearance or combustion space. To illustrate the latter point, suppose that, as is sometimes the case, the head of the cylinder is a flat plate. Then if the compression ratio be 6 : 1, as in this case, and the stroke equal to the diameter, the working fluid is compressed into a space of which the length is only one-fifth of the diameter, and the ratio of surface to volume extremely unfavourable. On the other hand, if the stroke were equal to five times the diameter, then the length and diameter of the clearance space would be equal, and the most favourable ratio of surface to volume would be obtained. Such a length of stroke would be quite impracticable for mechanical reasons, but the illustration serves to show the advantages of employing as long a stroke as is consistent with practical requirements.

Influence of Nature of Fuel on Power Output and Efficiency.—It has already been emphasized that the mean effective pressure depends primarily upon the weight of air that can be taken into the cylinder per cycle; since, other things being equal, the ultimate power output of an engine depends upon the weight of oxygen that can be burnt in a given time. The mean pressure

depends also upon the efficiency with which the fuel can be burnt, upon the heat of combustion, and is influenced by any change in the volume of the working fluid before and after combustion. In the case of some fuels, such as petrol and alcohol, the volume after combustion is some 5 to 6 per cent greater than before, whereas with ordinary illuminating gas it is some 3 per cent less. It is obvious that the change in specific volume has a considerable and direct influence on the mean effective pressure and efficiency.

The calorific value of the fuel affects the power output of an engine only in so far that when fuels of low calorific value are used, a considerably greater volume of fuel must be taken into the cylinder, and a similar volume of air is therefore displaced by it.

The heat of combustion, that is to say the total amount of heat liberated when all the carbon in the fuel has been converted into CO_2, and all the hydrogen to H_2O, has but little connection with the calorific value, and in fact the heat of combustion of all hydrocarbon fuels is substantially the same; it only varies appreciably when the fuel contains nitrogen or other inert diluents in addition to hydrogen and carbon.

On the other hand, the presence of diluents in the gas tends to raise its self-ignition temperature and pressure, and so permits of the use of a higher compression ratio, thus increasing the efficiency, and therefore, so far as the mean pressure is concerned, compensating to some extent for the reduced weight of air. Also, there is some evidence to indicate that the presence of diluents tends to increase the range of burning, and so permits of the use of weaker mixtures, and a lower maximum temperature, with the result that a still higher efficiency can be obtained, though, in this case, at the expense of power output.

The following table gives, column 1, the approximate heat of combustion in terms of B.T.U.s per cubic foot of working fluid (that is air and gas in the proportion required for complete combustion) of a number of fuels; other things being equal, this might be interpreted as the relative power output available with each fuel. Column 2 gives the approximate compression ratio permissible with each fuel, column 3 the efficiency corresponding to this compression ratio, and column 4 the mean effective pressure based on the following assumptions:

1. That the volumetric efficiency, expressed in terms of standard pressure and temperature, is 75 per cent in every case.
2. That there is no change in specific volume.

That the efficiency with which the fuel is burnt is, in all cases, 64 per cent of the air cycle efficiency for a mixture strength giving maximum power.

In column 5 is shown the mean effective pressures obtainable in practice after making approximate allowances in each case for—

1. The change in specific volume.

2. In the case of both petrol and benzol the volumetric efficiency will be somewhat higher, because the fuel is seldom completely evaporated before entering the cylinder, and some of the heat of the residual exhaust products is absorbed in overcoming the latent of evaporation of the fuel, with the result that the suction temperature is lower, and the weight of charge greater.

3. Owing to the wide variation in the heats of combustion and therefore of the flame temperature, the efficiency with which the fuel is burnt will vary to some extent; it will clearly be higher, for example, in the case of a blast-furnace gas, which consists mainly of inert diluents, than in the case of petrol or benzol, which contains no diluents at all.

TABLE

Fuel.	1 B.T.U.s per cu ft. of Mixture with Air.	2 Suitable Compression Ratio.	3 Efficiency per cent.	4 Mean Pressure lb. per sq. in.	5 Revised Mean Pressure lb. per sq. in.
Illuminating gas ...	89	6 : 1	32·8	118	110
Coke-producer gas ...	67·5	6·5 : 1	33·8	92·5	91·5
Anthracite-producer gas	68·5	6·5 : 1	33·8	94	93
Blast-furnace gas ..	59·5	7 : 1	34·5	83·5	85
Coke-oven gas	91	5 : 1	30·4	112	116
Petrol	99·5	5 : 1	30·4	122·5	138
Benzol	100	6·5 : 1	33·8	137	150

The above results, particularly those in column 5, must be regarded as approximations only, for very little accurate data is as yet available. They are, however, probably accurate to within ± 5 per cent.

CHAPTER IV

MECHANICAL EFFICIENCY

In all these calculations only the indicated horse-power has been considered. The conditions governing the mechanical efficiency have already been outlined, but will now be investigated in more detail in relation to this particular engine. As has been explained, these cannot always be reconciled with those required for maximum indicated efficiency, so that a compromise must be effected. It has already been shown that in a modern four-cycle engine the relative efficiency, that is, the ratio between the actual and ideal efficiencies, is about 86 per cent to 88 per cent, and the mechanical efficiency is about the same; in other words, the avoidable losses, both thermal and mechanical, are just about equal, and their recovery is therefore equally important. This is a point that must be borne in mind when deciding upon a compromise between thermal and mechanical requirements. Taking the typical engine in question, which will probably have a mechanical efficiency under full-load running conditions, of 87 per cent, then, as has already been shown, with a mixture strength of 10 per cent, the indicated horse-power will be 53·5 and the brake horse-power

$$\frac{87}{100} \times 53{\cdot}5 = 46{\cdot}5 \text{ B.H.P.}$$

Also, the indicated thermal efficiency is 35 per cent, and the brake or net thermal efficiency will be

$$\frac{87}{100} \times 35 \text{ per cent} = 30{\cdot}5 \text{ per cent.}$$

The mechanical losses, therefore, amount to 53·5 – 46·5, or 7 horse-power, and, under normal conditions, will be accounted for as follows:—

Pumping	1·9	horse-power.
Piston friction	3·6	"
Other friction (bearings, valves, &c.) ...	1·5	"
Total	7·0	"

From this it will be seen that piston friction forms by far the largest item and amounts to more than half of the total loss. A proportion of this loss is, in the author's opinion, preventible, and later in this volume the question of piston friction is discussed at some length. But, before proceeding further, it may be well to point out that there is good reason to believe that piston friction is mainly dependent upon, and due to, the inertia of the reciprocating masses, though there is, of course, a certain amount of constant friction due to the piston rings. During the first half of each stroke the crankshaft is doing work in accelerating the piston, and during the second half this work is returned to the crank, less a certain percentage lost in friction, which is more or less proportional to the inertia pressures. In this particular case, if the weight of the reciprocating parts be taken as 5·3 lb. per square inch of piston area, then the maximum pressure due to their inertia is approximately given by the formula

$$F = 0{\cdot}00017\, w\, n^2 s,$$

where w = weight of reciprocating parts per square inch of piston area,
n = revolutions per minute,
s = stroke in feet.

In this case, the value of F will be

$$F = 0{\cdot}00017 \times 5{\cdot}3 \times 240 \times 240 \times 1{\cdot}5$$
$$= 77{\cdot}8 \text{ lb. per square inch.}$$

The mean pressure per stroke, neglecting for the time being the angularity of the connecting-rod, which has no particular influence on the point in question, will be

$$\frac{77{\cdot}8}{2} = 38{\cdot}9 \text{ lb. per square inch.}$$

This is the mean pressure, in terms of pounds per square inch of piston area, due to the inertia of the reciprocating masses, and since it occurs throughout every stroke it must be multiplied by four in order to make it comparable to the useful mean fluid in the cylinder, which amounts in this case to 86·4 lb. per square inch. The total

pressures therefore acting upon the piston and tending to produce piston friction are:—

Useful fluid pressure	... ...	86·4 lb. per square inch.	
Pressure due to inertia (38·9 × 4)	...	155·6 „	„
Fluid pressure during idle strokes (exhaust, suction, and compression)	... ...	51·0 „	„
Total	... ...	293·0[1] „	„

From the above it will be seen that of the total pressures acting on the piston the mean positive fluid pressure amounts to only 30 per cent. and therefore a wide variation in the fluid pressure will have only a comparatively small influence on the piston friction. The proportion of the total pressure on the piston that is absorbed in friction will depend largely upon the temperature of the cylinder walls, which controls the viscosity of the lubricant, the area of piston in contact with the walls, and the efficiency of the lubrication. With normal temperatures and normal lubrication it has been found to amount to about 2 per cent. which is equivalent in this case to a mean pressure, when referred to the power stroke only, of 5·9 lb. per square inch of piston area, or 3·6 horse-power. The friction of the piston rings and that due to the fluid pressures during the idle strokes may be regarded as constant, and equal to approximately 1 lb. per square inch; this is practically unaffected by any change in the load or speed.

In support of the above distribution of the mechanical and fluid losses the author will quote the following published tests, which have been carried out with a view to determining this most important point. Unfortunately, in no case is the weight of the reciprocating parts recorded.

1. Professor Hopkinson, in his tests on a single-cylinder Crossley engine of 11·5 in. bore and 21 stroke, found that under normal conditions as to jacket temperature and lubrication the losses were as follows when the engine was indicating 41 I.H.P., its most economical load, and running at 180 R.P.M. = 630 ft. per minute piston speed:—

Fluid loss ...	1·4 horse-power	=	3·4 per cent of I.H.P.	
Piston friction	2·5 „	=	6·1 „	„
Other friction	1·1 „	=	2·7 „	„
Total ...	5·0 „	=	12·2 „	„

The mechanical efficiency therefore amounted to 87·8 per cent,

[1] The fluid and inertia pressures acting upon the piston are not, of course, always cumulative; their actual relation will be explained later, but for the purposes of the present argument they may be regarded as being cumulative without affecting the conclusions to any material degree.

and the piston friction accounted for just one-half of the total fluid and friction losses. A further and more detailed survey of this important question is given in another Chapter.

2. Mr. Herbert Chase, in his tests of a six-cylinder Pierce Arrow petrol engine in the laboratory of the Automobile Club of America, obtained the following results.

The engine had six cylinders, each $4\frac{1}{8}$ in. bore and $5\frac{1}{2}$ in. stroke, and, judging from the size and lift of its valves and the design of the engine generally, the normal piston speed of this engine should be about 1100 ft. per minute, corresponding to a rotative speed of 1200 R.P.M. At this speed the following results were attained:—

Indicated power	... ...	73·5 horse-power.
Brake horse-power	... ...	64·0 „
Mechanical efficiency	... ...	87·1 per cent.

Very careful tests were made to ascertain the mechanical and fluid losses, and these, with normal jacket temperature and lubrication, amounted to:

Fluid loss ...	3·4 horse-power	=	4·6 per cent of I.H.P.	
Piston friction	4·7 „	=	6·4 „	„
Other friction	1·4 „	=	1·9 „	„
Total ...	9·5 „	=	12·9 „	„

Here again the piston friction amounts to almost exactly half the total losses. In this case both the piston and rotative speeds are much higher than in Professor Hopkinson's tests, but the size of the valves and weight of the reciprocating parts are proportioned for the higher speeds.

3. Mr. L. G. Morse, in his tests on an old pattern of Daimler engine in the laboratories of Cambridge University, obtained the following results.

This engine had four cylinders, each of 3·56 in. bore and 5·11 in. stroke, and was intended for a normal speed of 1000 R.P.M., equal to a piston speed of 850 ft. per minute. Tests, however, were carried out at speeds of 720, 1000, and 1220 R.P.M., and the following results were obtained.

The losses in relation to the indicated horse-power at the three different speeds were found to be as follows:—

R.P.M.	720	1000	1220
Fluid loss ...	2·9 per cent	3·8 per cent	6·8 per cent.
Total friction ...	5·7 „	9·2 „	11·4 „
Total ...	8·6 „	13·0 „	18·2 „

The mechanical efficiency amounted therefore to 91·4 per cent at 720 R.P.M., 87 per cent at 1000 R.P.M., 81·8 per cent at 1220 R.P.M. Unfortunately, no attempt was made to separate the piston friction from the other friction losses; but if it be assumed that the other sources of friction bear the same proportion to the total as in the Pierce Arrow engine, namely, 1·9 per cent in each case, then the piston friction amounts to 3·8 per cent, 7·3 per cent, and 9·5 per cent respectively. The high fluid losses at 1220 R.P.M. are to be accounted for by the fact that the valves of this engine were not large enough for so high a speed.

Losses other than Piston Friction.—Returning now to our original engine, the pumping losses, which are not merely mechanical, and will in future be referred to as fluid losses, amount to 1·9 horse-power, or rather less than 4 per cent of the total indicated power. This is the loss incurred in expelling the products of combustion and drawing in the fresh charge. From both the mechanical and thermal points of view it is desirable that this loss should be reduced to a minimum. Its reduction can only be brought about by the provision of large valves, and careful design of the pipework generally, in order to avoid sudden changes of velocity, and take advantage, where possible, of the inertia forces of the gases in the exhaust and inlet pipes.

The third item, namely, bearing friction and the power required to operate the valves, amounts to about 3 per cent of the total indicated horse-power, and is probably not susceptible of any great reduction. The provision of an ample supply of lubricant, forced through the bearings under pressure, will effect a slight reduction in the loss from this source, and the adoption in high-speed engines of ball bearings for the crankshaft seems to be a step in the right direction. Such bearings are perfectly reliable, but the principal objection to their use is that they are apt to be somewhat noisy under intermittent loads. Bearing friction, however, is the smallest item of the three, and consequently it is not worth while devoting much time or ingenuity over it. It is of far greater importance to ensure that the bearings have ample surface and are not overloaded.

From the above conditions it is evident that very little reduction can be made in the fluid and bearing friction losses, and that attention should be concentrated upon the piston friction.

Influence of Weight of Reciprocating Parts.—Before dealing with the question of piston friction and piston design in

detail, it will be well to investigate the effect of varying either the load or the speed of the engine, upon the mechanical efficiency. It may, however, be broadly stated here that the piston friction depends upon the weight of the reciprocating parts, and that if the speed be varied, it increases approximately as the square of the speed of rotation.

Supposing, firstly, that by some means, such as air-scavenging, supercharging, &c., the mean effective pressure could be increased by (say) 40 per cent without any increase in the maximum temperature, then the piston and bearing friction losses will not be appreciably affected, for it may be accepted that the mean effective pressure is only a small item among the conditions producing piston friction. Then if the fluid losses remain unchanged, the total losses will, in this case, amount to 7·4 horse-power. The indicated horse-power will become 75, and the brake horse-power 67·6. The mechanical efficiency has now risen to

$$\frac{67 \cdot 6}{75} = 90 \text{ per cent.}$$

If the indicated thermal efficiency be the same in both cases, then the net efficiency becomes

$$\frac{90}{100} \times 35 \text{ per cent} = 31 \cdot 5 \text{ per cent.}$$

That is to say, although the thermal conditions in the cylinder have remained unaltered, the increase in the mean pressure has had the effect of increasing the net efficiency from 30·5 per cent to 31·5 per cent, an increase of about 3·5 per cent in the efficiency and of 45 per cent in the power. It is hardly conceivable, however, that the weight of charge could be increased by 40 per cent without increasing the fluid losses very appreciably, but even so there will probably be a gain in both the thermal and mechanical efficiency, in addition to the very large and valuable increase in the brake horse-power.

Secondly, supposing that the mean effective pressure remained the same, and that the speed of rotation were increased, then both the fluid and piston friction will increase more or less as the square of the speed, and the consequent losses as the cube of the speed. The bearing friction, however, will vary directly as the speed.

Effect of Increased Revolutions.—If the rotational speed

be raised from 240 R.P.M. to 360 R.P.M., that is in the ratio 1·0 : 1·5, then the losses will be approximately as follows:—

Fluid	... $1{\cdot}9 \times 1{\cdot}5^3$	=	6·4 horse-power.
Piston friction	... $3{\cdot}6 \times 1{\cdot}5^3$	=	12·2 „
Bearing friction	... $1{\cdot}5 \times 1{\cdot}5$	=	2·25 „
Total ...	... ...	=	20·85 „

The increase in indicated horse-power, other things being equal, will be directly proportional to the speed, and will in this case be

$$53{\cdot}5 \times 1{\cdot}5 = 80{\cdot}2 \text{ I.H.P.}$$

The brake horse power now becomes

$$80{\cdot}2 - 20{\cdot}85 = 59{\cdot}35,$$

and the mechanical efficiency,

$$\frac{59{\cdot}35}{80{\cdot}2} = 74 \text{ per cent.}$$

With a 50-per-cent increase of speed, the loss of heat to the cylinder walls will be reduced, but by no means in proportion, because the higher speed will involve greater turbulence in the gases in the cylinder, and therefore the absolute rate of heat flow will be greater. If the loss of heat to the cylinder walls be 12 per cent at normal speed, with an increase of speed of 50 per cent, this loss will probably drop to about 10 per cent, and the indicated thermal efficiency will be raised to nearly 36 per cent. In support of this the author would refer to Mr. Morse's tests on the thermal and mechanical efficiency of a petrol engine. Mr. Morse found that the indicated thermal efficiency, with a given strength of mixture, increased from 25 per cent at a piston speed of 720 ft. per minute to 27·7 per cent at 1080 ft. per minute. This engine, being very small, had a much greater ratio of surface to volume in the combustion chamber, and, consequently, the proportionate heat loss during combustion and explosion was far greater; but in the typical gas engine under discussion, the author considers that the increase in indicated thermal efficiency, with an increase of 50 per cent. in the piston speed quoted above, is not an over-estimate. The net thermal efficiency now becomes

$$\frac{74}{100} \times 36 \text{ per cent} = 26{\cdot}6 \text{ per cent.}$$

There will also be a further loss due to vibration, for since this is a single-cylinder engine, and therefore unbalanced, the disturbing forces tending to set up vibration will be increased as the square of the speed, and will probably become excessive.

A glance at the above figures is sufficient to show that it would be impracticable to increase the speed of this engine as it stands to 360 R.P.M. The increase in brake horse-power would be dearly paid for by a much reduced net efficiency and excessive vibration. The brake horse-power would, however, be slightly higher than 59·3, because the reduction in the heat loss will produce a corresponding increase in the mean effective pressure. Against this, however, must be offset the fact that the valves are too small to deal with the extra volume of working fluid, and there will therefore be considerable throttling, which will reduce the mean effective pressure to a degree that will probably much more than counteract the thermal gain.

Supposing now that the engine were redesigned with a view to running at 360 R.P.M., then, to obtain the same mechanical efficiency, the weight of the reciprocating parts would have to be reduced in the ratio of 1 to $1·5^2$; that is, to 44 per cent of their original weight, or 2·36 lb. per square inch of piston. The valves and pipework will have to be enlarged, and so designed that, even at 360 R.P.M., the velocity of the gases when passing through the valve ports is not greater than 130 ft. per second. Since the velocity of the piston at 360 R.P.M. is 1080 ft. per minute, or 18 ft. per second, it follows that the ratio between the area of the valves and that of the piston must be as 18 : 130. The effective area of opening of a poppet valve is equal to the area of the valve port when the lift is equal to one-quarter of the diameter of the port. The ratio, therefore, between the diameter of the valve ports and that of the piston would have to be as

$$\sqrt{18} : \sqrt{130} = 4·23 : 11·4.$$

Since the diameter of the piston is 12 in., the diameter of the valves must be not less than

$$\frac{4·23}{11·4} \times 12 = 4·5.$$

In practice it will be preferable to use (say) 5-in.-diameter valves and reduce the lift to 1 in.

Valves of this size are perfectly practicable, and there will not

be the smallest difficulty in obtaining an equally good volumetric efficiency at the higher speed, while the fluid loss will be no more than proportional to the speed. Under the revised conditions the horse-power absorbed in overcoming the frictional and fluid losses will be as follows:—

Fluid loss ...	..	$1·9 \times 1·5 =$	2·85 horse-power.
Piston friction	...	$3·6 \times 1·5 =$	5·4 ,,
Bearing friction	...	$1·5 \times 1·5 =$	2·25 ,,
Total	...	...	10·50 ,,

Since the volumetric efficiency will be at least as high as before, and since the mean pressure will be increased by about 3 per cent due to the reduction in the heat loss, the indicated horse-power will now be

$$53·5 \times 1·5 \times \frac{103·5}{100} = 83 \text{ I.H.P.},$$

and the indicated thermal efficiency 36 per cent.

The brake horse-power now becomes

$$83 - 10·5 = 72·5.$$

The mechanical efficiency is

$$\frac{72·5}{83} = 87·4 \text{ per cent},$$

and the brake thermal efficiency

$$36 \text{ per cent} \times 87·4 \text{ per cent} = 31·5 \text{ per cent}.$$

By reducing the weight of the reciprocating parts from 5·3 lb. to 2·36 lb., the inertia forces, and consequently the load on the bearings, will be exactly the same as with the heavier reciprocating masses at the lower speed.

From the above calculations it will be seen that if the necessary reduction could be made in the weight of the reciprocating parts, the brake horse-power could be increased from 46·5 to 72·5, and the net thermal efficiency from 30·5 per cent to 31·5 per cent, without any increase in the vibration or in the load on the bearings. Apart from the gain in thermal efficiency, the value of the engine will be increased in proportion to the brake horse-power, in this case in the proportion of 1·56 to 1. The necessary enlargement of the valves and pipework will not introduce any serious mechanical difficulties, or have an appreciable effect upon the cost, so that there is a

considerable sum of money available to cover the extra cost of lighter reciprocating parts.

It must, however, be borne in mind that the total amount of heat that passes into the cylinder walls and piston has been increased nearly in the ratio of 12 : 10 × 1·5, or 25 per cent; for the rate of heat loss to the cylinder walls depends very largely upon the velocity of the gases in the cylinder, and this again depends upon the rotative speed. In larger engines this would be a serious objection, but in an engine of 12-in. bore or less, very little trouble need be anticipated on this score.

As a practical example of what can be accomplished by scientific study, and the reduction of the reciprocating weights, the author would refer his readers to the tests recently carried out by the German War Office authorities on a four-cylinder Benz aeroplane engine, and published in 1913. This engine runs normally at a piston speed of no less than 1520 ft. per minute, and a rotative speed of 1288 R.P.M. The net thermal efficiency is 29[1] per cent. Since the fuel used is petrol, which has a comparatively low ignition temperature, the compression ratio could not well be higher than 5 to 1, so that the air standard efficiency for this engine is only 47·5 per cent. If the relative efficiency be taken as 70 per cent, which is considerably above the highest figure that the author has ever seen obtained by any normal engine of this kind, then the indicated thermal efficiency is 33·2 per cent, and since the net efficiency is 29 per cent, it follows that the mechanical efficiency must be over 87·4 per cent.

More remarkable still, this high thermal efficiency is obtained with an indicated mean pressure of over 120 lb. per square inch, showing that at these very high speeds the heat loss to the cylinder walls must be very small, and that an excellent volumetric efficiency can be obtained.

Effect of Longer Stroke.—Returning again to the original design, suppose that, instead of increasing the rotative speed, this were still kept at 240 R.P.M., but that the piston speed were again raised to 1080 ft. per minute, by a 50-per-cent increase in the stroke. Then, since the volume of working fluid to be dealt with per minute has been increased, it will be necessary to increase the size of the valves and passages as before, and the fluid losses may

[1] A number of tests were carried out on this engine during 1914 both by the Admiralty and the War Office. These tests, which are all in fair agreement, give the net thermal efficiency as 27 per cent, the compression ratio being about 4·9 : 1.

again be taken as 2·85 horse-power. The bearing friction will also increase directly as the piston speed, because, although the rotary speed of the bearings has not been altered, it will be necessary to increase their diameter nearly in proportion to the stroke, and the actual rubbing velocities will therefore be approximately proportional to the piston speed. For this reason the bearing friction may again be taken as 2·25 horse-power. Taking next the piston friction, it is clear that the inertia of the reciprocating parts varies directly as the piston speed, if the rotative speed be kept constant. Consequently, if the original heavy reciprocating parts be retained, the piston friction will be increased in the same proportion as the piston speed, namely, by 50 per cent, and will amount to 5·4 horse-power.

The total losses in this case will be

Fluid loss ...	...	...	2·85	horse-power.
Piston friction	...	...	5·4	„
Bearings ...	...	...	2·25	„
Total	...	...	10·50	„

The increase in piston speed will, as previously explained, be accompanied by a slight increase in the indicated thermal efficiency, and therefore in the indicated horse-power, also the smaller ratio of surface to volume in the combustion space will help in this direction. Under these circumstances an indicated thermal efficiency of 36 per cent may be expected, and the mean pressure will be increased by about 3 per cent, owing to the reduced heat loss.

The indicated horse-power will therefore be

$$55·5 \times 1·5 \times \frac{103·5}{100} = 83 \text{ I.H.P.}$$

The losses, as already shown, will amount to 10·5 horse-power, so that the brake horse-power will be

$$83 - 10·5 = 72·5.$$

The mechanical efficiency will be

$$\frac{83}{72·5} = 87·4 \text{ per cent.}$$

The brake or net thermal efficiency will be 31·5 per cent.

From the above figures it will be seen that if, instead of reducing the weight of the reciprocating parts and increasing the rotative speed, the stroke be increased by 50 per cent, with the same weight

of reciprocating parts and the same rotative speed, the same results will be obtained, but in this case a 50-per-cent increase in the stroke will involve a nearly similar increase in the weight, bulk, and cost of the engine. Still better results will, of course, be obtainable if the reciprocating weights be reduced, but it must be remembered that the longer-stroke engine requires a proportionately longer, and therefore more than proportionately heavier, connecting-rod.

From a practical commercial point of view it will be preferable to increase the rotative speed rather than the stroke, and it will be found profitable to go to a considerable amount of trouble and expense over the reduction of the reciprocating weights. There is, however, a strong prejudice amongst engineers against any increase in the rotative speeds, on the grounds that such increase must be accompanied by a loss of mechanical efficiency, excessive wear, and vibration. But this, as has been shown, can be met by a reduction in the reciprocating weights. The prejudice, nevertheless, still remains, and will not easily be dissipated.

Value of Light Reciprocating Parts.—For real progress in this direction one must turn to the modern petrol engine, and especially to those engines built for racing motor-cars. Of late years all motor-car races have been run under restricted conditions either as to piston area or swept volume. From about 1905 till 1910 practically all motor-car engines were handicapped according to the area of their pistons, and there was a mistaken impression prevalent at that time that it was the piston and not the rotative speed that was limited; i.e. that a short-stroke engine would run at a proportionately higher maximum speed. Manufacturers of racing cars, however, soon discovered how erroneous this view was, and produced engines of small bore and excessive length of stroke, which by their higher piston speeds soon swept all before them. After a few years of racing under these conditions it became evident that a type of engine was being developed which, owing to its excessive stroke-bore ratio, and therefore its high cost and increased vibration, was of little commercial value. At the present time nearly all motor races are handicapped according to the swept volume of the cylinders, and under these conditions a far more rational type of engine is being produced. Since no restrictions whatever are placed on fuel consumption, the designers of racing petrol engines have concentrated all their energies upon improving the mechanical and volumetric efficiencies of their designs, the thermal efficiency being considered only in so far as it affects the mean pressures.

Nevertheless, it is surprising what remarkably high thermal efficiencies are occasionally obtained from these engines, while the lessons learnt regarding volumetric and mechanical efficiencies are of inestimable value.

As an illustration of the results obtained from a modern racing petrol engine, the following figures, obtained by Professor Riedler from a 100-horse-power Benz racing car engine, are taken from his book, *The Scientific Determination of the Merits of Automobiles.*

The leading dimensions of this engine were as follows:—

Number of cylinders	4.
Bore	115 mm., or 4·5 in.
Stroke	175 mm., or 6·9 in.
Rotative speed	2000 R.P.M.
Piston speed	2303 ft. per minute.
Compression ratio (r)	4·75 : 1.
Area of piston	15·9 sq. in.
Total volume of each cylinder	139 cu. in.
Volume swept by piston	109·7 cu. in.
Clearance volume	29·3 cu. in.
Ratio of swept volume to total volume ...	3·75 : 1.
Weight of reciprocating parts	5·75 lb.
Weight of reciprocating parts per square inch of piston area	0·36 lb.
Area of exhaust-valve opening	3 sq. in.
Area of inlet-valve opening	4·4 sq. in.
Ratio piston area to inlet area	3·62 : 1.

The brake horse-power of this engine was found to be

55 B.H.P. at 1000 R.P.M.,
103·5 B.H.P. at 2000 R.P.M.

The indicated horse-power was

60·5 I.H.P. at 1000 R.P.M.,
119 I.H.P. at 2000 R.P.M.

This corresponds to a mechanical efficiency of

91 per cent at 1000 R.P.M.,
and 87 per cent at 2000 R.P.M.,

which is certainly a most remarkable result.

The indicated mean pressures were

109·5 lb. per square inch at 1000 R.P.M.,
107·5 „ „ 2000 „

The mean velocity through the inlet valves was

$$\frac{3{\cdot}62 \times 1150}{60} \text{ ft. per second} = 69{\cdot}4 \text{ ft. per second at } 1000 \text{ R.P.M.,}$$
$$\text{and } 138{\cdot}8 \text{ ft. per second at } 2000 \text{ R.P.M.}$$

The exhaust valves, however, were considerably smaller, and, reckoned on the same basis, the mean velocity through them was

$$\frac{15{\cdot}9 \times 1150}{60} \text{ ft. per second} \doteq 101{\cdot}5 \text{ ft. per second at } 1000 \text{ R.P.M.,}$$
$$\text{and } 203 \text{ ft. per second at } 2000 \text{ R.P.M.}$$

The latter figure is distinctly high, but in this engine the connecting-rods were exceptionally short, with the result that there was a comparatively long "dwell" at or near the bottom centre.

The weight of the reciprocating parts is remarkably low. This result was secured by the sacrifice of wearing surface and by the use of very short connecting-rods, which would not be permissible in the case of a stationary engine required to run for long periods without undue wear. It is worthy of note that Professor Riedler comments on the fact that when run continuously at full load, and at a speed of 2000 R.P.M., the piston crown overheated, and gave rise to pre-ignition, thus indicating that weight cutting had in this instance been carried too far.

Calculating the mechanical efficiency by the method to be explained later, the fluid or pumping losses when running at 2000 R.P.M. will be equivalent to a mean pressure of approximately 3·5 lb. per square inch on the piston.

The friction of the bearings and other parts will be exceedingly low, and will probably be equal to a mean pressure of not more than 2 lb. per square inch, for the power of the engine is enormous when compared with the weight of the rotating parts.

The piston friction may be found as before. The useful fluid pressure amounts to 107·5 lb. per square inch. The mean inertia pressure amounts to

$$F = 0{\cdot}00017 \times 0{\cdot}36 \times 2000 \times 2000 \times \frac{6{\cdot}9}{12} \times 0{\cdot}5$$
$$\doteq 70{\cdot}5 \text{ lb. per square inch,}$$

or $70{\cdot}5 \times 4 = 282$ lb. per square inch when referred to the power stroke only.

The total pressure, therefore, acting on the piston amounts to

$$282 + 107\cdot 5 = 389\cdot 5 \text{ lb. per square inch.}$$

The constant friction due to the piston rings and the fluid pressure during the idle strokes may be taken as equal to 1·5 lb. per square inch, and the friction due to the inertia and useful fluid pressure as 2 per cent of the total pressure, or

$$389\cdot 5 \times 0\cdot 02 = 7\cdot 79 \text{ lb. per square inch.}$$

The total piston friction therefore amounts to 9·3 lb. per square inch.

The total losses due to pumping, bearing, and piston friction will therefore amount to

Fluid loss	3·5 lb. per square inch.		
Piston friction	9·3	,,	,,
Bearings and other friction ...	2·0	,,	,,
Total losses ...	14·8	,,	,,

The indicated mean pressure at 2000 R.P.M. was found to be 107·5 lb. per square inch.

Hence the brake mean pressure will be

$$107\cdot 5 - 14\cdot 8 = 92\cdot 7 \text{ lb. per square inch,}$$

corresponding to a mechanical efficiency of

$$\frac{92\cdot 7}{107\cdot 5} = 86\cdot 2 \text{ per cent,}$$

which agrees very closely with the figures obtained by Dr. Riedler.

The volumetric and thermal efficiencies are not given, but it is clear from the high mean pressures obtained that these must both be very high. In an Adler racing engine of somewhat similar proportions, but with rather smaller proportionate valve areas, which gave similar results, the volumetric efficiency was found to be about 73 per cent at a speed of 2000 R.P.M., and 80 per cent at 1000 R.P.M. The drop in mean effective pressure, however, between these two limits of speed was not nearly in proportion to the drop in the volumetric efficiency, showing that the proportion of heat lost to the cylinder walls during the expansion stroke was considerably less at the higher speed. This should result in a higher thermal efficiency at the higher speed, as was indeed the case.

The indicated thermal efficiency was found to be

at 1000 R.P.M., 26 per cent,
at 2000 R.P.M., 30·5 per cent.

The difference seems too great to be accounted for by the reduction of heat loss alone, and it is probable that the engine received either a slightly weaker or perhaps more homogeneous mixture at the higher speed.

The air standard efficiency for this particular engine is 46·7 per cent, and the relative efficiencies are therefore

at 1000 R.P.M., 55·7 per cent,
at 2000 R.P.M., 65·5 per cent.

It is evident from the above figures that an increase of piston speed may be relied upon to produce a marked increase in the thermal efficiency, provided, of course, that the valve area is sufficient, and the reciprocating parts sufficiently light to ensure that the whole advantage gained is not neutralized by excessive piston friction.

The above figures serve to illustrate the fact that the designers of high-speed petrol engines in 1911 were able to produce engines which, at the extremely high rotative speed of 2000 R.P.M., and at a piston speed of no less than 2300 ft. per minute, could show almost as high a volumetric, and a higher mechanical efficiency than the designers of horizontal gas engines generally obtained with a rotative speed of 240 R.P.M. and a piston speed of only 720 ft. per minute. The comparison, however, is not quite a fair one, because in the case of the petrol engine the bearing surfaces of the piston were not sufficient for long continuous running. But, even when due allowance has been made for this, the petrol engine is still vastly superior from the point of view of mechanical efficiency. Still further progress has been made during the last few years with high-speed petrol engines, and the maximum power of the latest racing engines is generally developed at about 3000 R.P.M., which, with a stroke of 6 in., as is now usual, gives a piston speed of 3000 ft. per minute.

CHAPTER V

CONDITIONS UNDER REDUCED LOADS

Systems of Governing.—Up to the present only the indicated thermal efficiency on full load has been considered, but all engines are required to run at times on a reduced load, and in a great many instances the engine is required to run normally on a light load, the full load being called for only on rare occasions. In such cases it is equally, or perhaps more important, that a high thermal efficiency shall be obtained on the lighter loads. With engines of the constant-pressure or Diesel type the load is regulated simply by the duration of the supply of fuel per stroke, and, as has been shown previously, the indicated thermal efficiency is considerably greater with light loads when the fuel is cut off very early in the stroke, and the maximum temperature is low. Unfortunately, however, it would appear that similar conditions cannot be obtained in an explosion engine. The methods which have been employed up to the present for controlling the output, i.e. for "governing" internal-combustion engines, are:—

1. Hit and miss.
2. Quantitative governing.
3. Qualitative governing.

The first method used to be a very popular one, and has much to recommend it for small engines. In this method separate valves are employed for the admission of gas and air. The air valve is opened at every suction stroke in the usual manner, but the gas valve is under the control of the governor, and is opened only when the load on the engine requires it. The quantity of gas taken in per stroke, and therefore the strength of the mixture, is the same as when running on full load, but under these conditions the indicated thermal efficiency and the mean effective pressure are slightly higher, for the following reasons:—

1. Because, following a missfire, the cylinder at the commence-

ment of the suction stroke contains pure air at a low temperature, instead of exhaust gases at a high temperature. Consequently, at the end of the suction stroke, on account of the lower temperature, which in this case will be about 130° F., the weight of mixture taken in will be greater.

2. Since the weight of air present in the clearance space is, on account of its lower temperature, much greater than the weight of exhaust gases that would be present under normal running conditions, it follows that the final mixture in the cylinder is slightly weaker in proportion, and the efficiency slightly higher.

3. Since the initial temperature is lower by 90° F., the temperature throughout the cycle will be correspondingly reduced, involving slightly less heat loss to the cylinder walls. The mean pressure will depend mainly upon the volumetric efficiency of the engine, and this, in turn, will depend upon the suction temperature. Thus, if the suction temperature be reduced by 90° F., the volumetric efficiency will increase in the proportion of

$$\frac{459 + 220}{459 + 130} = \frac{1 \cdot 15}{1}, \text{ or an increase of 15 per cent.}$$

The increase in mean pressure will also be nearly 15 per cent, so that when running with a 10-per-cent mixture of gas and air the actual mean pressure will be 86·4 lb. per square inch on full load, and about 100 lb. per square inch after a missfire; actually it will be even higher than this, owing to the small increase in efficiency.

The principal advantages in favour of the *hit-and-miss* system of governing are: 1. That the indicated thermal efficiency of the firing stroke is slightly greater on light loads than on full load. 2. The system allows of exceedingly close and accurate governing, and at the same time puts practically no load upon the governor. 3. The variations in the load are effected without changing the strength of the mixture, so that, when once adjusted, it remains the same for all loads, which is an important practical consideration.

The principal disadvantage, and one which puts the system altogether out of court for larger engines, is that on light loads the turning moment is altogether too irregular, necessitating a fly-wheel of such dimensions as to be quite impracticable; for small engines, in cases where good cyclical regularity is of little importance, it is, in the author's opinion, the most convenient system.

The second system, namely, *quantitative governing*, has the very marked advantage in that it provides an impulse in every cycle at

all loads, and the cyclical regularity is excellent, a matter of considerable importance when the engine is used for such purposes as direct electric lighting. In this system, in order to reduce the power, the supply of both gas and air is cut down either by means of a throttle valve actuated by the governor, or by reducing the lift of the main inlet valve, which, in this case, supplies both gas and air. The effect of reducing the weight of charge taken into the cylinder is, of course, to reduce the mean effective pressure proportionately; but since the quantity of exhaust products retained in the combustion space is nearly the same at all loads, it follows that the proportion of exhaust products to fresh mixture increases as the load is diminished. This is objectionable for three reasons: firstly, because it increases the temperature of the mixed gases in the cylinder at the commencement of the compression stroke; secondly, because the proportion of exhaust products in the final mixture is so great, and the mixture so diluted that it is necessary to use a richer mixture in order to obtain sufficiently rapid combustion; thirdly, the reduction in mean pressure is effected by reducing the weight and not the temperature of the charge. All these conditions tend to reduce the thermal efficiency. It is not worth while investigating this question in great detail, because, in practice, the real controlling factor is the partially incomplete combustion caused by the dilution with exhaust products, and this, which probably has the greatest influence of all upon the efficiency, is an uncertain quantity.

In order as far as possible to obviate such dilution, attempts are made to separate the fresh charge from the exhaust products by means of stratification, and it is a common practice to arrange the inlet valve so that, instead of opening into the cylinder, it opens into a pocket in which the igniter is fitted, as shown in the engine illustrated. In this way, the fresh charge entering through the inlet valve first enters the pocket and remains at this end of the cylinder, while the exhaust products are more or less concentrated over the piston. This arrangement involves very little increase in the surface of the combustion chamber, while the extra efficiency on light loads, obtained from the more certain and complete combustion, more than compensates for the extra heat losses due to the increase of exposed surface; at all events if the engine is required to run on light loads for a large proportion of the time.

Quantitative governing is invariably employed for four-cycle petrol engines used for such purposes as propelling motor cars, boats, &c., where a very wide range of speed and power is required,

and where good cyclical regularity is of paramount importance. It is also generally used for large engines employed for such purposes as generating electricity; indeed, it is almost the only system that can be relied upon to ensure regular firing over a wide range of load.

Qualitative governing consists in varying the percentage of gas in the mixture. The quantity of air taken in at all times remains the same, but the quantity of gas is varied to suit the load. Since the range of mixture strength over which complete combustion can be relied upon is comparatively small, it follows that qualitative governing can only be effective between comparatively narrow limits, and that the efficiency is at a maximum at the lightest load which will ensure complete combustion, for, as has already been shown, the thermal efficiency falls as the proportion of gas to air is increased. Some fuels have a wider range over which complete combustion will take place than others, but in any case the range is far too small to permit of any wide variation of power. In order to increase the range of power as far as possible, every effort is made to encourage stratification. The admission of the working fluid is arranged progressively, air only entering the cylinder during the first portion of the suction stroke. The gas enters with the last portion of the air, so that the main body of the combustion space at the end of the compression stroke contains almost pure air, while the pocket containing the inlet valve and igniter contains a readily combustible mixture of gas and air. This, as has been previously explained, is the condition required for the maximum of thermal efficiency, and it is the ideal that the designer of an engine employing qualitative governing strives after. The success so far attained with four-cycle engines, however, has not been very encouraging, although designers of two-cycle engines, who have to rely on qualitative governing, have developed the possibilities of stratification to a much higher pitch. Mention should here be made of the very ingenious valve gear employed by Messrs. Crossley & Co., on their qualitative governed engines, which is described in detail in another volume; this valve evidently produces a fair degree of stratification.

There can be little doubt that when more is known of the phenomena of stratification, and the conditions which control it, qualitative governing will be more extensively employed, for no other system can compare with it in efficiency. At the present time, however, it has hardly been sufficiently developed to be considered as a commercial success, chiefly on account of the

danger of "firing back". If, as is commonly the case on very light loads, the combustible mixture, through defective stratification, is excessively diluted by the large excess of pure air in the cylinder, the charge either fails to ignite at all and is therefore lost, or, what is even worse, ignites and burns so slowly that combustion is still continuing throughout the whole of the exhaust stroke. The result is that when the inlet valve opens, the entering charge comes in contact with the still burning gases, and ignites back through the valve and into the valve passages, making a considerable noise and fouling the succeeding charges. This is an all too common occurrence with qualitative-governed engines, and is particularly noticeable in two-cycle engines. Crossley's Patent Inlet Valve is designed with a view to preventing such back-firing by admitting only pure air at first, but since, at the end of the preceding suction stroke, the valve was supplying a combustible mixture of gas and air, there is always a slight danger of a small portion of this mixture being retained behind the valve and drawn into the cylinder ahead of the pure-air charge.

Both qualitative and quantitative governing rely to a greater or lesser extent upon stratification for light loads, and for this reason it is advisable to fit the inlet valve and igniter in a recess or pocket in the combustion chamber. This, of course, increases the area of surface exposed to combustion and so increases the heat loss, but it is thoroughly justified by the better and more complete combustion obtained on light loads. In the case of hit-and-miss governing this consideration does not apply, and the shape of the combustion space is a compromise between the thermodynamic and the mechanical requirements. For horizontal, single-acting engines, the position of valves and shape of combustion space illustrated in the typical example, fig. 11, is almost invariably employed, whether the governing be by hit-and-miss, quantitative, or qualitative. The arrangement gives a perceptible degree of stratification, with a comparatively small proportion of exposed surface, while the valves are readily accessible and easily operated from a single side shaft.

Mechanical Efficiency on Reduced Load.—Considering next the mechanical efficiency of this typical engine when running on a reduced load. Suppose that the load be reduced to one-third of the normal full load, that is to 15·5 B.H.P. The reduction of load may be effected by missing explosions, by throttling, or by qualitative governing.

Taking first the case when power is reduced by missing ex-

plosions. It has already been shown that, owing to the scavenging action, and absence of highly-heated exhaust gases in the clearance space, the mean effective pressure during an expansion stroke, following a scavenging stroke, is generally some 15 per cent higher. In this engine, with a 10-per-cent gas-air mixture, the normal mean pressure is 86·5 lb. per square inch, but following a scavenging stroke, it will probably be about 100 lb. per square inch; also, owing to the lower temperatures ruling in the cylinder, the efficiency will be slightly higher, say 36 per cent.

The fluid losses will be considerably increased, and Professor Hopkinson has found that the losses in pumping during the scavenging strokes are about 2·5 times as great as under normal conditions. The reasons for this are not at first sight obvious, and it is worth while examining them in detail. Firstly, during the suction stroke, a slightly greater volume of air is taken into the cylinder, and hence proportionately greater power is absorbed. Secondly, during the compression and expansion strokes heat is lost to the cylinder walls, and consequently the expansion line is well below the compression line; that is to say, the energy absorbed in compressing the air is not all returned during the expansion stroke, but a small proportion of it passes as heat to the cylinder walls. The loss on this account alone will amount to about 1·4 horse-power. Thirdly, the power required to expel the air during the exhaust stroke is much greater, for there is no high pressure in the cylinder at the time when the exhaust valve is first opened to create a high velocity in the exhaust pipe and so help to withdraw the remaining contents of the cylinder. The diagram, fig. 16., illustrates these points very clearly. The dotted line is the diagram obtained during the idle strokes when the engine is running under normal full-load conditions, and the full line shows that obtained during the same strokes when scavenging.

For these reasons the fluid losses during the scavenging period may be taken as 4·75 horse-power as against 1·9 horse-power when firing.

Considering next the piston friction, since the rotative speed is the same in both cases, the load due to the inertia of the reciprocating parts will be the same, namely 155·6 lb. per square inch. When the total mean pressure on the piston, referred to the expansion stroke, amounts to 245·5 lb. per square inch, the horse-power lost in friction is 3·6 horse-power. When the fluid pressures are removed, the mean pressures on the piston will be only that

due to the inertia, and will amount, as has already been shown, to 155·6 lb. per square inch, for the mean pressure during the idle strokes is provided by the constant = 1 lb. per square inch.

The piston friction when the engine is not firing now becomes

$$\left(155{\cdot}6 \times \frac{2}{100}\right) + 1 \text{ lb. per square inch}$$
$$\doteq 4{\cdot}1 \text{ lb. per square inch, or } 2{\cdot}5 \text{ horse-power.}$$

The losses due to bearing friction and valve operation depend

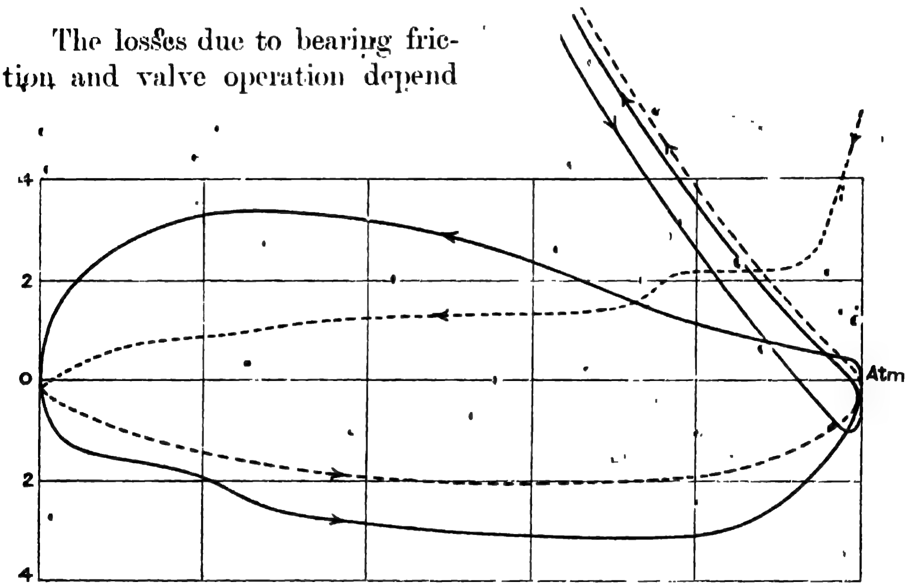


Fig 16.—Light Spring Diagram

mainly upon the speed and the inertia forces, and will therefore not be affected.

The losses during the scavenging strokes may, therefore, be taken as follows:—

Fluid losses ...	...	...	4·75	horse-power.
Piston friction	...	...	2·5	"
Other friction	...	...	1·5	"
Total ...	...	...	8·75	"

When firing, the lost horse-power will be 7·0 horse-power as before, and since at this load the engine will be firing approximately 38 per cent and missing approximately 62 per cent of the cycles, the average loss will be

$$\frac{(8{\cdot}75 \times 62) + (7{\cdot}0 \times 38)}{100} = \frac{542 + 266}{100}$$
$$\doteq 8{\cdot}1.$$

Since the brake horse-power required is 15·5 B.H.P., it follows that the indicated horse-power must be

$$15{\cdot}5 + 8{\cdot}1 = 23{\cdot}6.$$

The mechanical efficiency will now be

$$\frac{15{\cdot}5}{23{\cdot}6} = 65{\cdot}7 \text{ per cent;}$$

and taking the indicated thermal efficiency as 36 per cent, the brake thermal efficiency will be

$$36 \times \frac{65{\cdot}7}{100} = 23{\cdot}6 \text{ per cent.}$$

In order to find the exact number of power strokes or impulses per minute required to give this power it will be necessary to find the indicated horse-power of each impulse stroke. This will be

$$\frac{100 \times 113{\cdot}1 \times 1{\cdot}5}{33000} = 0{\cdot}515,$$

and if the I.H.P. required be 23·6 the number of impulses per minute will be

$$\frac{23{\cdot}6}{0{\cdot}515} = 46 \text{ impulses per minute.}$$

If, instead of missing explosions, the power be reduced by throttling the incoming charge, so that ignition takes place at every cycle, but at a very much reduced mean pressure, then the thermal efficiency becomes an unknown quantity, because with very much reduced charges the combustion is seldom anywhere near complete. Also, it is generally necessary to use a somewhat richer mixture than at full load, in order to ensure ignition of the charge. Experience shows that the full-load indicated thermal efficiency is generally about 75 to 80 per cent of the full-load efficiency. Taking the same brake horse-power as before, namely 15·5, it can be shown that the mean effective pressure will be slightly under 40 lb. per square inch, and consequently the weight of charge taken in per cycle will be only about 46 per cent of that taken in on full-load. This will involve a considerable suction loop in the indicator diagram, and hence a large fluid loss during the suction stroke. This is well illustrated by the indicator diagrams, fig. 17 (see Plate facing p. 80), which are actual diagrams, taken with an optical indicator, from the author's experimental engine running on a light load and throttle-governed, but

under special conditions ensuring rapid and complete combustion. The area of the suction loop, and therefore the fluid losses, could be reduced by reducing the period of opening of the inlet valve, instead of by throttling, but this is seldom done in comparatively small engines. On the other hand, there will be no fluid loss due to the compression and expansion of unburnt gases, and no extra fluid loss on the exhaust stroke, since the engine is firing every cycle. Without going into elaborate figures, it will probably be safe to take the fluid loss in this case at about 4·5 horse-power.

The piston friction may be taken as

$$\left((155{\cdot}6 + 40) \times \frac{2}{100}\right) + 1 \text{ lb. per square inch}$$
$$= 5 \text{ lb. per square inch, or } 3{\cdot}05 \text{ horse-power.}$$

Bearing friction and valve-operating losses may be taken as being the same in all cases.

The losses will now be as follows:—

Fluid loss	4·5	horse-power.
Piston friction	3·05	"
Other friction losses ...	1·5	"
Total friction ...	9·05	"

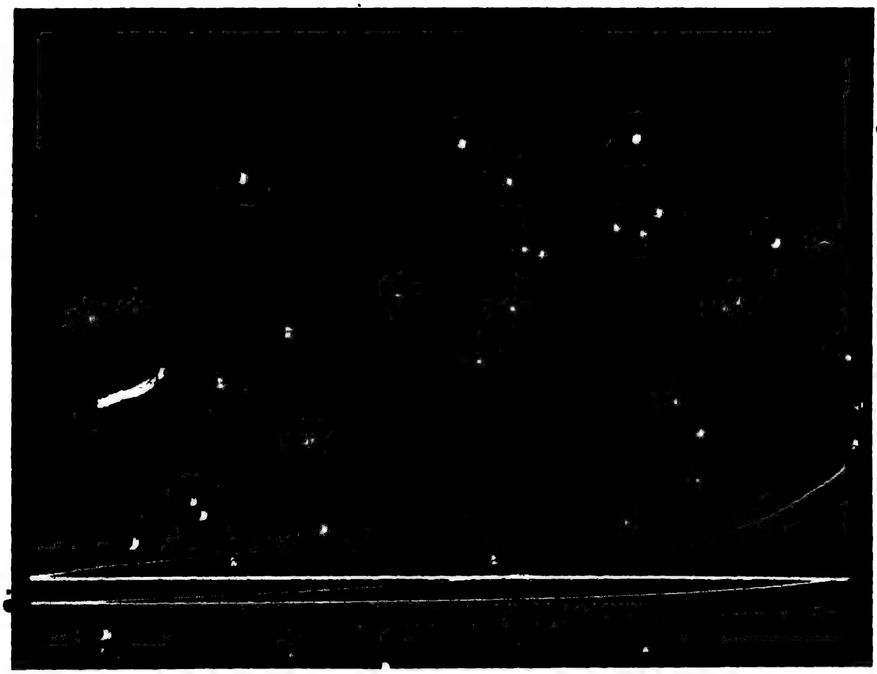
If the brake horse-power required be 15·5, then the indicated horse-power will be

$$15{\cdot}5 + 9{\cdot}05 \text{ I.H.P.} = 24{\cdot}55 \text{ I.H.P.}$$

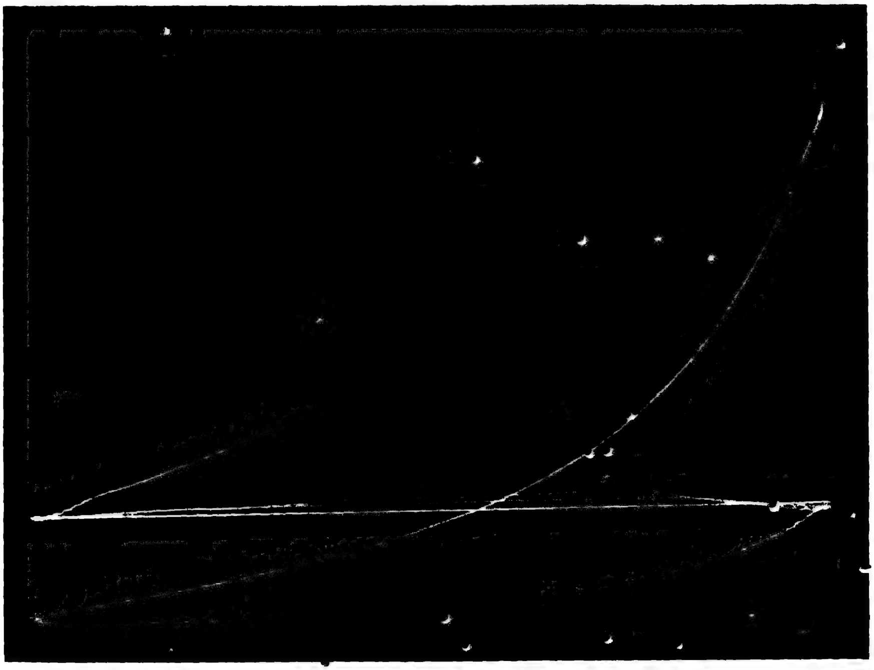
The mechanical efficiency will be

$$\frac{15{\cdot}5}{24{\cdot}55} = 63{\cdot}2 \text{ per cent.}$$

Supposing combustion to be complete, and the mixture strength exactly the same as on full load, then the heat lost to the cylinder walls would be nearly the same in both cases. The differences are due to two, among other, causes—1. Since the weight of charge taken into the cylinder is very much less, there will be a lower entering velocity and less turbulence. 2. Since the proportion of exhaust gases present in the cylinder is greater, the maximum temperatures will be slightly lower. For these reasons less total heat will be lost to the cylinder walls on a reduced load. On the other hand, since the weight of charge taken in is only about 45 per cent of the full-load weight, the proportionate loss at light loads will be much greater. Taking all these points into



Indicator Spring 100 lb. = 1 inch



Indicator Spring 20 lb. = 1 inch

Fig. 17

consideration, it is probable that the proportionate heat loss to the cylinder walls will be somewhere in the neighbourhood of 18 per cent.

The indicated thermal efficiency will probably be about 28 per cent, and the brake thermal efficiency

$$63 \cdot 2 \times 28 \text{ per cent} = 17 \cdot 7 \text{ per cent.}$$

But it has already been pointed out that there will probably be a further loss due to incomplete combustion, and that in consequence the brake thermal efficiency will be lower.

Finally, assuming that the load were reduced by reducing the density of the mixture, that is to say, by qualitative governing. To obtain 15·5 B.H.P. a mixture density of somewhere in the neighbourhood of 4 per cent will be required, and such a mixture, if homogeneous, would be altogether too weak to ignite. Consequently stratification must be relied upon, in order that the mixture immediately surrounding the igniter points may be of a sufficient density to ensure rapid and complete combustion. In the light of present knowledge such a condition is generally accepted as being unattainable, but it is none the less interesting to consider the case on the assumption that it is capable of being attained.

Taking the fluid and mechanical losses; since the engine is firing at every cycle, and is taking in a full charge of air without throttling, the fluid losses will be approximately the same as on normal full load, namely, 1·9 horse-power. The piston friction will be the same as when running with throttle governing, namely, 3·05 horse-power, and the bearing and other friction losses may be taken as being the same in all three cases.

The losses will now be as follows:—

Fluid loss	1·9	horse-power.
Piston friction	3·05	,,
Other friction losses ...	1·5	,,
Total	6·45	,,

The indicated horse-power will therefore be

$$15 \cdot 5 + 6 \cdot 45 = 21 \cdot 95,$$

and the mechanical efficiency

$$\frac{15 \cdot 5}{21 \cdot 95} = 70 \cdot 6 \text{ per cent.}$$

Now the indicated thermal efficiency in this case will be the same as for a 4-per-cent mixture, and, as will be seen from the curve, fig. 13, will be about 44 per cent. Hence the brake thermal efficiency under these conditions will be

$$\frac{70 \cdot 6}{100} \times \cdot 44 = \cdot 31.$$

In practice a brake thermal efficiency of 31 per cent at one-third load has never been obtained, or even approached, in an

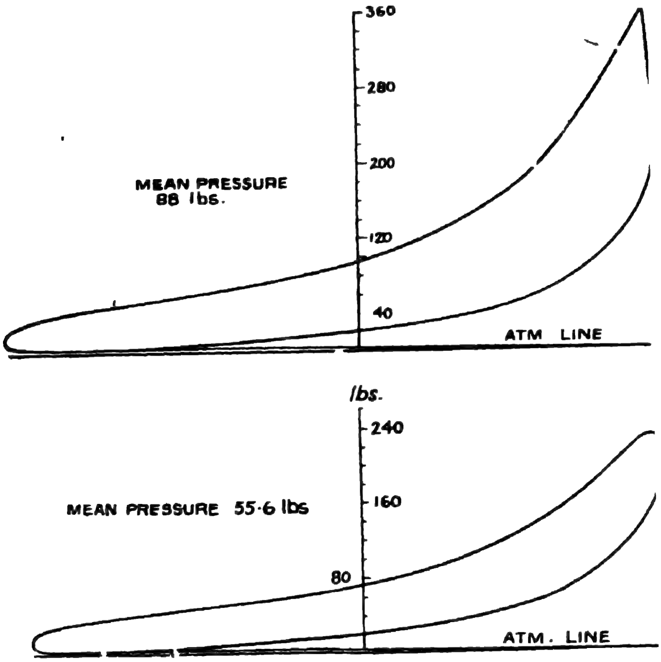


Fig. 18

engine of this type, but the example is interesting as showing what might be accomplished if perfect stratification could be realized. The diagrams illustrated in fig. 18 are taken from an engine employing qualitative governing at full and at half load. The half-load diagram illustrates very clearly the reduction in the maximum temperature and pressure, which should result in increased efficiency, but it also illustrates slow and incomplete combustion, and it is probable, from an examination of the diagram, that the thermal efficiency actually obtained on reduced loads is little if any higher than when governed by throttling the charge.

CHAPTER VI

THE DIESEL ENGINE

The typical gas-engine which has just been analysed may be regarded as being representative of all modern four-cycle constant-volume or explosion engines. Larger engines are generally built with double-acting cylinders, and, of course, external cross-heads. In this case the piston friction is considerably reduced and a higher mechanical efficiency is obtained. It is not proposed at this stage to deal with the two-cycle explosion engine, in which the problems connected with the thermal and mechanical efficiencies are very much more complicated, and which are dealt with at some length in Vol. II, but rather to pass on to the constant-pressure or Diesel type.

A typical example of a Diesel engine is illustrated in figs. 19 and 20. It will be noted that this is a vertical design, which is usually adopted for constant-pressure engines, not because there is any particular reason why one should be built with horizontal and the other with vertical cylinders, but simply because the Diesel type was developed at a later period in the history of the Internal-combustion Engine, when vertical cylinders were becoming the accepted type in contemporary steam-engine practice.

It has already been shown that in the constant-pressure engine the theoretical efficiency depends not only upon the compression ratio, but also upon the maximum temperature, and increases as the maximum temperature is reduced. In this particular instance, for the sake of comparison, it will be assumed that on full load the admission of fuel is continued for a sufficient length of time to produce the same maximum temperature as in the gas-engine. In order to pulverize the fuel sufficiently finely to ensure complete combustion, it has been found desirable to force it into the cylinder by means of highly compressed air, at a pressure considerably in excess of that in the cylinder. The quantity of air used for the

injection of the fuel usually amounts to about 6 per cent of the air taken into the cylinder, and this somewhat confuses the issue. Let it be supposed that the engine has the following leading dimensions:—

Bore	12 in.
Stroke	18 in.
Rotative speed	240 R.P.M.
Compression ratio	14 : 1.
Area of piston	113·1 sq. in.
Piston speed	720 ft. per minute.
Proportion of injection air to total air	6 per cent.
Pressure of injection air	900 lb. per sq. in. on full load.
Weight of reciprocating parts ...	800 lb.
Weight of reciprocating parts per square inch of piston area ...	7·1 lb.

The weight of the reciprocating parts in this engine has been taken as 800 lb. The reason for the increase in the weight of these parts, as compared with the gas-engine, is that they must be sufficiently strong and heavy to be able to resist not only the normal working pressures but also any abnormal pressures that may occur. In the gas-engine with a 6 : 1 compression ratio the highest pressure that is likely to occur in the event of pre-ignition with a strong mixture is under 650 lb. per square inch; but in the constant-pressure engine very much higher pressures may occur under certain abnormal conditions, and it is advisable so to design all parts of the engine that they can be relied upon safely to withstand a pressure of from 1200 to 1500 lb. per square inch.

Indicated Efficiency.—Taking first the indicated efficiency and power. It is not proposed to go through the same lengthy calculation as in the case of the constant-volume engine, which would be particularly laborious in this case, because allowances would have to be made for the fact that heat is added at constant pressure and rejected at constant volume; and also for the fact that air, which has been previously compressed from some external source, is admitted to the cylinder along with the fuel during the expansion stroke.

It has already been shown that the theoretical air-standard efficiency of a constant-pressure engine is dependent both upon the compression ratio and upon the maximum temperature. We have also seen that the efficiency rises as the maximum temperature is reduced, until, at the point of no-heat supply, it is equal to the

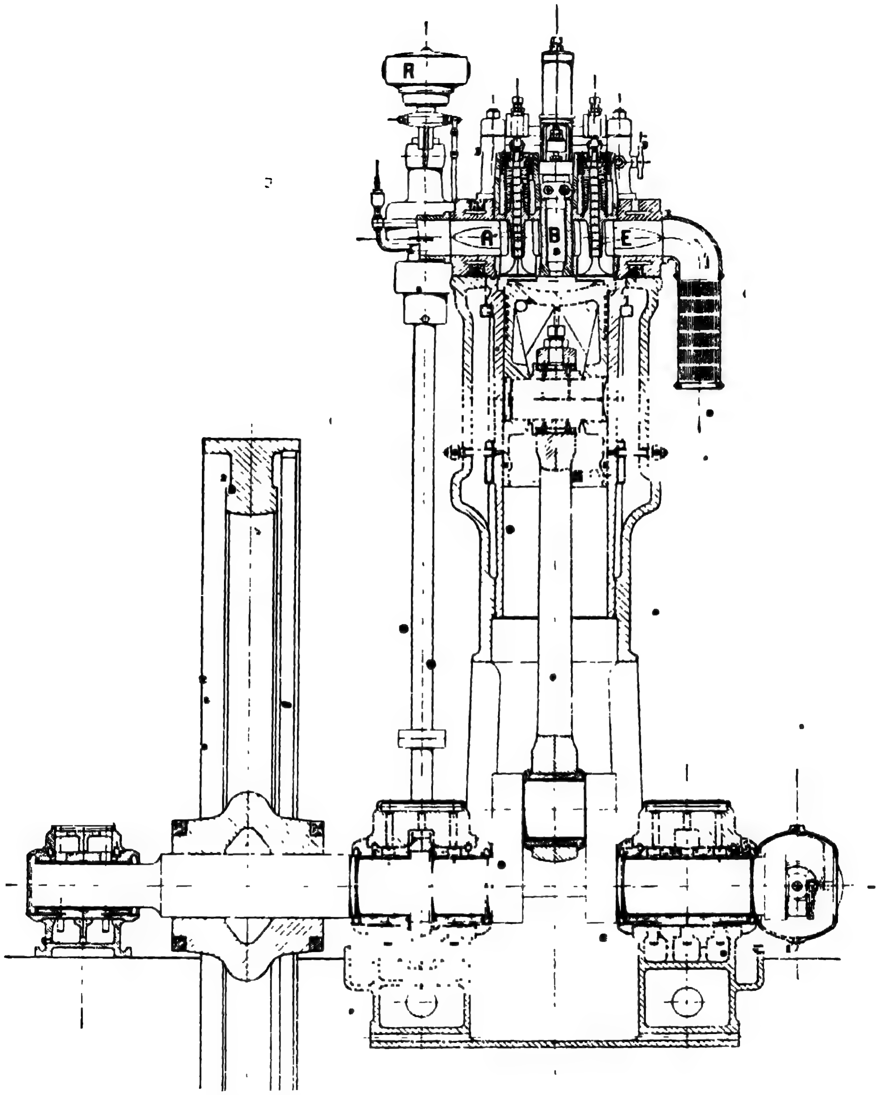


Fig. 19 – Section of a Typical Four-cycle Diesel Engine

air-standard efficiency for a constant-volume or explosion engine, which in this case amounts to

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1}$$

$$= 1 - \left(\frac{1}{14}\right)^{0.4}$$

$$\doteq 65 \text{ per cent.}$$

Under normal full-load working conditions, with a maximum temperature of 3480° F., the theoretical air-standard efficiency will be about 56 per cent, but this will be still further reduced when the changes in specific heat have been taken into account.

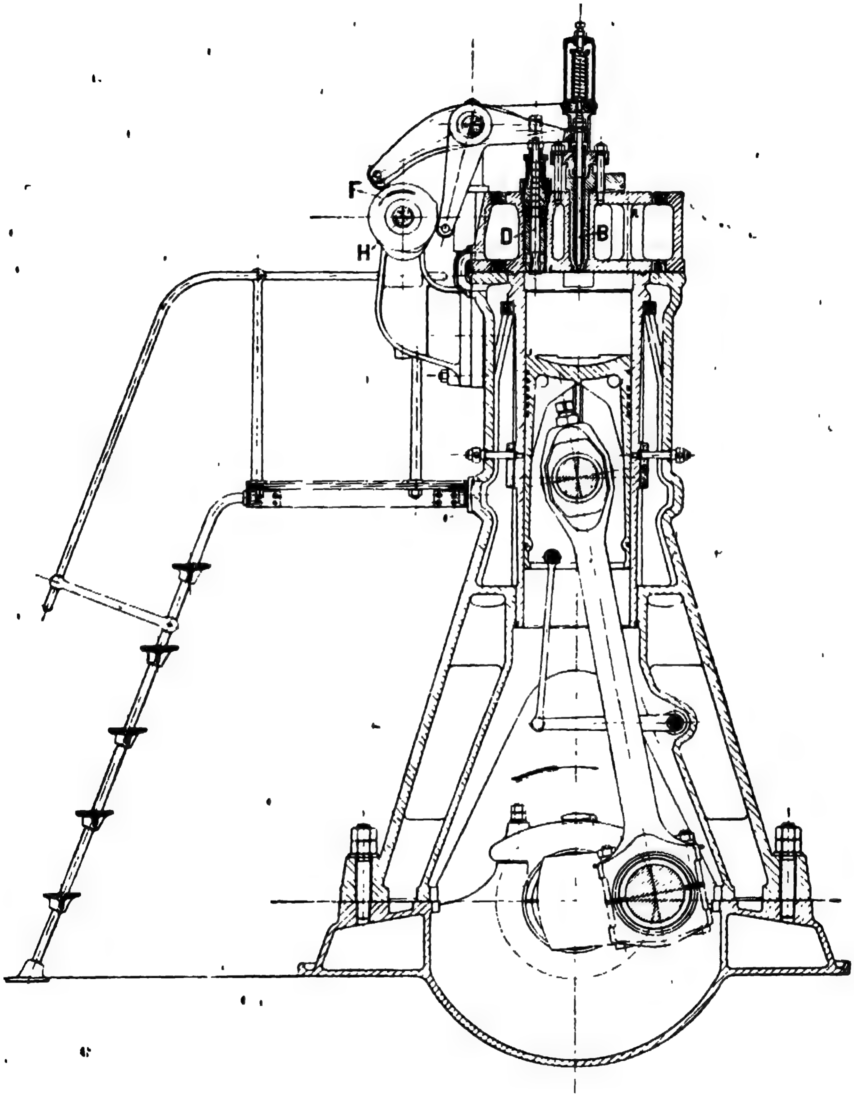


Fig. 20. - Section of a Typical Four-cycle Diesel Engine

On the other hand, the introduction of an extra supply of air after the end of the compression stroke will tend to raise the apparent efficiency somewhat. For the purposes of the following calculations it will be assumed that, with a maximum temperature

of 3480° F., the ideal efficiency of this engine is 51 per cent. It will also be assumed that the efficiency rises in a straight line as the maximum temperature, and therefore the mean effective pressure, is reduced until it finally meets the air-standard efficiency line at the point of no-heat supply. In this engine the apparent volumetric efficiency will be higher than in the gas-engine, because about 6 per cent of extra air is added during the combustion period. Again, since the specific heat of air at constant pressure is about 40 per cent higher than the specific heat at constant volume, about 40 per cent more heat is required to heat the gas through a given interval of temperature at constant pressure than is required when the heating is done at constant volume. Taking all these points into consideration, the ideal mean effective pressure, corresponding to a maximum temperature of 3480° F., may be taken as 135 lb. per square inch, as against 100·5 lb. per square inch in the gas-engine.

In actual practice, with a maximum temperature of 3480° F., and making allowance for the extra air admitted, an indicated thermal efficiency of about 43 per cent may be obtained, and under these conditions the indicated mean pressure will be

$$\frac{43}{51} \quad 135 = 114 \text{ lb. per square inch,}$$

and the indicated horse-power

$$\frac{114 \times 113 \cdot 1 \times 1 \cdot 5 \times 120}{33000} = 70 \cdot 5 \text{ I.H.P.}$$

Mechanical Efficiency.—Taking the indicated horse-power as 70·5 I.H.P. it now remains to find the mechanical efficiency and brake horse-power, and to this end it is first necessary to investigate the various sources of loss.

The fluid loss will be approximately the same as in the gas-engine, and may therefore be taken as 1·9 horse-power. On account both of the heavier reciprocating parts and the higher mean pressure the piston friction will necessarily be greater. That portion of the friction due to the inertia of the reciprocating parts will vary directly as the weight of these parts, and, when reduced to terms of pressure per square inch of piston per power stroke, will become

$$\frac{7 \cdot 1}{5 \cdot 3} \quad 155 \cdot 6 = 208 \text{ lb. per square inch.}$$

Add to this the useful mean pressure, amounting to 114 lb. per

square inch, and the total pressures tending to produce piston friction will amount to

$$208 + 114 = 322 \text{ lb. per square inch.}$$

The constant friction loss due to the piston rings and the fluid pressures during the idle strokes must also be added, and in this case may be taken as amounting to 2 lb. per square inch, for the higher compression will necessitate more and stiffer rings and greater pressure during the compression stroke.

The piston friction will therefore be

$$\left(322 \times \frac{2}{100}\right) + 2 \text{ lb. per square inch}$$

$$= 8{\cdot}44 \text{ lb. per square inch, or } 5{\cdot}15 \text{ horse-power.}$$

The bearing friction in this engine will be somewhat higher than in the gas-engine, because, to withstand the higher mean pressures, larger bearing surfaces are necessary; the crankshaft also must be of considerably larger diameter, with the result that the bearing friction will probably be about 30 per cent greater.

Finally, the valve gear will require more power, because, in addition to the inlet and exhaust valves, there is also a fuel valve, which is only opened for an exceedingly short time, and which requires a very stiff spring to ensure rapid closing. Moreover, as a general rule, it has to be opened against the injection air pressure. There is also a fuel pump, which will absorb an appreciable amount of power, since the oil has to be delivered to the fuel valve against the pressure of the injection air. It will probably not be overstating the case to put the bearing and valve gear friction at 3·5 horse-power.

In addition to these losses, there is also the high-pressure compressor, used for supplying the injection air for the fuel valve. The power absorbed by this compressor varies considerably, according to the amount of air required for pulverization, which again depends upon the design of the fuel valve, the load, the speed, and the form of the combustion chamber.

Under the conditions of load and speed in this particular case, and with the best possible design of air-pump, delivering 6 per cent of the air for combustion through the fuel valve, the power absorbed will not be less than 7 horse-power, and in many cases will be considerably more.

The total mechanical and fluid losses are now as follows:—

Fluid loss	...	..	...	1·9 horse-power.
Piston friction	...	..		5·1 „
Other friction	...	..		3·5 „
Air compressor	...	...		7·0 „
Total	...	...		17·5 „

The brake horse-power will therefore be

$$70{\cdot}5 - 17{\cdot}5 = 53 \text{ B.H.P.}$$

The mechanical efficiency will be

$$\frac{53}{70{\cdot}5} = 75 \text{ per cent,}$$

and the brake thermal efficiency

$$43 \times \frac{75}{100} = 32{\cdot}2 \text{ per cent.}$$

If the lower heating value of residual oil be taken as 18300 B.T.U.s per pound, this will correspond to a fuel consumption of

$$0{\cdot}43 \text{ lb. per B.H.P. hour.}$$

Comparing these latter results with those obtained from a constant-volume gas-engine of the same dimensions, running at the same speed, and employing the same maximum temperatures, it will be seen that the brake horse-power and the brake thermal efficiency are both only very slightly higher, though a study of the respective compression ratios would lead one to expect a much higher net efficiency. In the gas-engine, with a compression ratio of 6 : 1, the air-standard efficiency is 51 per cent, and the actual net efficiency 30·5 per cent, or within 60 per cent of the air standard. In the constant-pressure oil-engine, with a compression ratio of 14 : 1 and an air-standard efficiency of 65 per cent, the net efficiency is only 32·2 per cent, or slightly less than 50 per cent of the air-standard efficiency. The principal causes of this large discrepancy are to be found in the reduction of the air-standard efficiency with increase of temperature, and of course the still further reduction when the changes of the specific heat of the working fluid at high temperatures are taken into account. There is also the formidable proportion of the total power required to drive the high-pressure air compressor for the pulverization of the fuel to be considered.

If means could be found for pulverizing and distributing the fuel without the necessity of this high-pressure air, a great improvement would be made, but, so far, no system of mechanical pulverization has been sufficiently successful to increase the net efficiency; that is to say, the losses due to imperfect pulverization and distribution, and consequent incomplete combustion, have been found to be greater than those incurred by the air compressor.

From the above results it will be seen that, as regards the net thermal efficiency on full load, there is very little to be gained by the adoption of the constant-pressure type in preference to the constant-volume. The former type is very much more expensive, involves a large number of small parts which require extremely accurate workmanship, and cannot well be made on the tools employed in an ordinary engineering workshop. Again, since much greater pressures may, under abnormal circumstances, occur in the cylinder, it is necessary to build the whole engine considerably heavier than an explosion engine of similar power. There are, however, two very strong arguments in favour of the Diesel type of engine. The first is that practically any liquid hydrocarbon fuel can be burnt direct in the cylinder, without the necessity of first passing it through a gas producer, with all the losses that such an intermediate stage must incur. Secondly, since the fuel is not present in the cylinder during compression, and does not depend upon thorough mixture with the air during this period for complete combustion, qualitative governing, with its manifold advantages, can be employed for all loads without having to rely upon stratification. It is these two features that have brought about the widespread popularity of the Diesel oil-engine.

Governing.—The theoretical advantages to be obtained by qualitative governing have already been considered in the case of the gas-engine. In the Diesel engine the advantages gained are even more important, for not only are the losses reduced; but the air-cycle efficiency itself rises as the temperature is reduced. In the Diesel engine the maximum temperature is proportional to the quantity of fuel admitted, and the efficiency rises as the load is reduced at an even greater rate than in the gas-

Let us consider the case of this engine when running at one-third load of a brake horse-power of 17·0. The fluid losses will be substantially the same at all loads, since a full supply of air is always taken in, irrespective of the load, and ignition of the fuel

occurs at every cycle, hence these losses at 17·6 B.H.P. may again be taken as 1·9 horse-power.

The piston friction will be slightly reduced, on account of the reduced fluid pressure, which in this case will be about 60 lb. per square inch. This loss will therefore amount to about

$$\left((208 + 60) \times \frac{2}{100}\right) + 2 = 7 \cdot 36 \text{ lb. per square inch,}$$

or 4·5 horse-power.

The bearing and other friction losses may be taken to be the same as before, namely, 3·5 horse-power. On account of the reduction in the quantity of fuel there may also be a reduction in the amount and pressure of the air required for its pulverization, and consequently the power absorbed by the air compressor will be less. In this case it may fairly be taken as 4·5 horse-power, instead of 7 horse-power, as on full load.

The total of the mechanical and fluid losses now becomes

Fluid loss ...	...	...	1·9	horse-power.
Piston friction	...	...	4·5	"
Other friction	...	...	3·5	"
Air compressor	...	...	4·5	"
Total losses		...	14·4	"

Since the brake horse-power is now 17·6 B.H.P., it follows that the I.H.P. necessary is

$$17 \cdot 6 + 14 \cdot 4 = 32 \text{ I.H.P.,}$$

and the mechanical efficiency is

$$\frac{17 \cdot 6}{32 \cdot 0} = 55 \text{ per cent.}$$

At this load the actual indicated thermal efficiency, making no reduction for the extra air supplied by the compressor, will probably be about 48 per cent, and the brake thermal efficiency will be

$$48 \times \frac{55}{100} = 26 \cdot 4 \text{ per cent.}$$

Corresponding to a fuel consumption of 0·528 lb. per B.H.P. hour.

At about two-thirds full load—that is, at a load of 36 B.H.P.—the mechanical and net thermal efficiencies will be approximately as follows:—

The mechanical and fluid losses will be

Fluid loss	1·9
Piston friction ...	4·9
Other friction ...	3·5
Air compressor ...	6·2
Total losses	16·5

Since the brake horse-power is now 36, it follows that the I.H.P. necessary is

$$36 + 16{\cdot}5 = 52{\cdot}5 \text{ I.H.P.}$$

The mechanical efficiency is

$$\frac{36}{52{\cdot}5} = 68{\cdot}5 \text{ per cent.}$$

At this load an indicated thermal efficiency of about 46 per cent may be expected, and the net thermal efficiency will therefore be

$$46 \times \frac{68{\cdot}5}{100} \qquad {\cdot}5 \text{ per cent.}$$

Corresponding to a fuel consumption of 0·44 lb. per B.H.P. hour. This method of computing the indicated thermal and mechanical efficiencies is one that is usually accepted, and, indeed, it is probably the only practicable method. The results obtained, however, are erroneous from a scientific point of view, because a certain proportion of extra air is admitted to, and does work in, the cylinder of the engine, but the work previously done on this air is included in the mechanical losses, instead of being deducted from the indicated power. The result of this computation is that the indicated thermal efficiency is too high and the mechanical efficiency too low. It is not suggested that any other computation should be adopted, nor, indeed, is it a matter of any consequence, for it is the overall efficiency alone which is of importance to the practical engineer. But it is well to bear in mind, when statements are made as to the indicated thermal efficiency of Diesel engines, that a comparison between the actual indicator diagram and the quantity of fuel consumed will give a figure for the indicated thermal efficiency which is considerably above the true value. As a comparative figure it is quite sufficiently accurate, but as an absolute one it is misleading, and therefore the efficiencies of constant-pressure and explosion engines should be compared only on their net values.

CHAPTER VII

TWO-CYCLE DIESEL ENGINE

Two-cycle Diesel Engine.—In the two-cycle Diesel engine the thermal conditions are practically the same as in the four-cycle engine, but the mechanical efficiency is considerably lower. In engines working on this cycle, since there is no suction or exhaust stroke, it is necessary to force the air for combustion into the cylinder at a pressure above the atmospheric pressure, and for this purpose a separate pump of some form must be employed. Supposing that the same typical constant-pressure engine were now converted to operate upon the two-stroke instead of the four-stroke cycle, it is interesting to compare the mechanical and net thermal efficiencies which would be obtained.

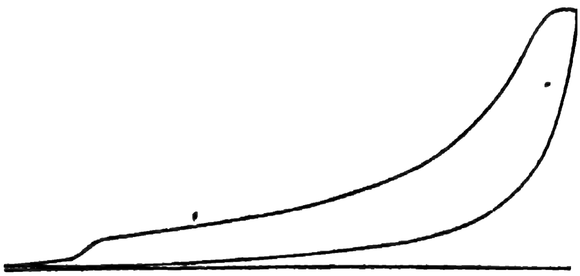


Fig. 21.—Two-cycle

In so far as the thermal conditions within the cylinder are concerned there will be very little change, but the effective stroke of the engine will be slightly reduced, because the last 20 per cent of the stroke will be completed after the exhaust ports have been uncovered, and the pressure within the cylinder has been released. In fig. 21 is shown an actual diagram from a two-cycle Diesel engine. Strictly speaking, the compression and expansion strokes should be regarded as beginning and ending at the point where the pressure in the cylinder crosses the atmospheric line, but this point is not easily determined with any precision, but on the average from 10 to 15 per cent of the expansion stroke may be regarded as lost. Consequently the mean pressure of two-cycle engines is for an equal maximum temperature always somewhat lower.

In order to drive out as much as possible of the exhaust gases

and thoroughly fill the cylinder with air, it is usual to employ a scavenging pump which displaces from 40 to 60 per cent more air than the swept volume of the cylinder. As a general rule, even with a pump capacity 60 per cent in excess of the cylinder volume, the proportion of air retained in the cylinder seldom exceeds 75 per cent of that available in a four-cycle engine. This proportion is sufficient for a mean pressure of about 110 lb. per square inch (reckoned on the full stroke). Such high mean pressures as this are, however, seldom employed, on account of the high temperatures involved, and these generally prove to be the limiting factor in all but small engines.

We will assume that the typical engine, already illustrated, is converted to operate on the two-stroke cycle, and that all the leading dimensions are unaltered. Assuming that it is running on full load with a mean pressure of 100 lb. per square inch, then since there is an impulse or power stroke at every revolution, the indicated horse-power will be

$$\frac{100 \times 113\cdot1 \times 1\cdot5 \times 240}{33000} = 123\cdot4 \text{ I.H.P.},$$

and the indicated thermal efficiency will be in the neighbourhood of 42 per cent, reckoned on the usual basis. For although on the one hand the maximum temperature is lower, from 10 to 15 per cent of the available expansion is lost, due to the piston overrunning the exhaust ports.

For purposes of comparison, it is assumed that the engine will run at the same speed whether it is operating on the two- or four-stroke cycles, but it should be remembered that, in the former case, the time available, both for the expulsion of the exhaust from, and for the entry of air to the cylinder, is substantially reduced, and that this may, in practice, necessitate running at a somewhat lower speed.

Mechanical Efficiency of Two-cycle Engine.—It is now necessary to investigate the various mechanical and fluid losses. Taking first the scavenging pump, this has a capacity equal to, say, 1·5 times the swept volume of the power cylinder. To simplify the following calculations, let it be assumed that the scavenge pump is also 12 in. bore but has a stroke of 27 in., or 50 per cent more than the power cylinder. The pressure of air required to overcome the resistance of the valves or inlet ports and expel the products of combustion is generally about 3·5 lb. per square inch, and this may be taken as a fair average figure. The pump, there-

fore, must deliver its full volume of air against a pressure of 3·5 lb. per square inch above atmosphere. The mean pressure in the pump cylinder will probably be greater than this, depending upon the size and type of valves employed, but, however carefully these are designed, it is unlikely that the mean pressure, including the suction loop, will be less than 5 lb. per square inch.

The indicated horse-power of the scavenge pump will therefore be

$$\frac{5 \times 113 \cdot 1 \times 27 \times 240}{33000 \times 12} = 9 \cdot 2 \text{ I.H.P.}$$

The mechanical efficiency of such a pump will probably be about 75 per cent, so that the horse-power required to overcome the mechanical friction of the pump will be

$$9 \cdot 2 \doteq 3 \cdot 1 \text{ horse-power.}$$

The main piston friction will, of course, be considerably reduced in proportion to the total power, because, while the inertia pressure remains the same, the useful mean pressure is nearly doubled when reckoned on the same basis as in a four-cycle engine. The inertia pressure on this basis amounts to 208 lb. per square inch as before, but in this case about 50 per cent of the total inertia pressure is balanced by the compression pressure, so that the net figure may be taken as about 110 lb. per square inch.

$$110 \text{ lb. per square inch} + 200 \text{ lb. per square inch} = 310 \text{ lb. per square inch.}$$

The piston friction will therefore be

$$\left(310 \times \frac{2}{100}\right) + 2 \cdot 0 = 8 \cdot 2 \text{ lb. per square inch.}$$

This friction is equivalent to 5·0 horse-power.

The bearing and other friction losses will be approximately the same in both cases, namely about 3·0 horse-power.

The quantity of high-pressure blast air required will be rather less than double the quantity required for the four-cycle engine on account of the lower mean pressure, and the mechanical friction of the compressor will be proportionately lower on account of the larger size. On these grounds the total power absorbed by the air com-

pressor may be taken as 12 horse-power. The total mechanical and fluid losses will be as follows:—

Scavenge pump	Fluid loss	9·2	horse-power.
	Mechanical loss	3·1	"
	Piston friction	5·0	"
	Other friction	3·0	"
	Air compressor	12·0	"
	Total losses ...	32·3	"

Since the indicated horse-power is 123·4 I.H.P., the brake horse-power will be

$$123{\cdot}4 - 32{\cdot}3 = 91{\cdot}1 \text{ B.H.P.}$$

The mechanical efficiency will be

$$\frac{91{\cdot}2}{123{\cdot}5} = 74 \text{ per cent.}$$

If the indicated thermal efficiency be 42 per cent, the net thermal efficiency will be

$$74 \times 42 = 31{\cdot}1,$$

corresponding to a fuel consumption of 0·455 lb. per B.H.P. hour. In comparing these results with the four-cycle engine it must be pointed out that, in the case of the two-cycle engine, owing to the lower mean pressure, the indicated thermal efficiency is taken as 44 per cent as against 43 per cent in the four-cycle engine. Experience with actual engines has shown that, taking the best examples of both types, the net thermal efficiency of the four-cycle engine is generally about 5 per cent higher on full load. On one-third load the two-cycle engine shows up less favourably, especially if, as is generally the case, the full-load volume of the scavenge pump be utilized. In this connection it should be pointed out that only a certain small proportion of oxygen is required for complete combustion of the fuel on light loads, the remainder of the air or gases being simply inert. Under these conditions there is no particular object in expelling exhaust gases at the cost of a considerable load on the scavenge pump, and a better mechanical and net thermal efficiency would be obtained if the quantity of scavenge air were reduced, though, so far as the author is aware, this is never done in practice.

On one-third load the brake horse-power of the two-cycle engine

will be approximately 30 B.H.P., and the mechanical and fluid losses will be:—

Scavenge pump	Fluid loss ...	9·2	horse-power.
	Friction loss ...	3·1	"
	Piston friction ...	4·0	"
	Other friction ...	3·0	"
	Air compressor ...	8·0	"
	Total losses	27·3	"

The indicated horse-power becomes

$$30 + 27{\cdot}3 = 57{\cdot}3 \text{ I.H.P.}$$

The mechanical efficiency

$$\frac{30}{57{\cdot}3} = 52{\cdot}3 \text{ per cent.}$$

The indicated thermal efficiency at 57·3 I.H.P. will be about 48 per cent, and the net thermal efficiency

$$48 \times 52{\cdot}3 = 25{\cdot}1 \text{ per cent,}$$

corresponding to a fuel consumption of 0·565 lb. per B.H.P. hour. If now on one-third load the air scavenge were reduced to one-half the quantity at full load, the fluid loss in the scavenging pump would be reduced from 9·2 horse-power to probably about 3·5 horse-power, for the reduction in volume of air would naturally be accompanied by a reduction in the pressure needed and its velocity. Under these conditions the mechanical and fluid losses would be reduced from 27·3 to 21·6 horse-power.

The indicated horse-power required would be

$$30 + 21{\cdot}6 = 51{\cdot}6 \text{ I.H.P.,}$$

and the mechanical efficiency would be increased to

$$\frac{30}{51{\cdot}6} = 58{\cdot}2 \text{ per cent.}$$

On account of the lower I.H.P. the indicated thermal efficiency should be higher, but this advantage will be counteracted to some extent, because the greater quantity of exhaust gases retained in the cylinder will raise the temperature throughout the cycle and so increase the heat losses. The difference on either side will be trifling and the net result will probably be the same, therefore the

indicated thermal efficiency may be taken as 48 per cent in both cases.

The net thermal efficiency now becomes

$$58 \cdot 2 \times 48 = 27 \cdot 9 \text{ per cent,}$$

corresponding to a fuel consumption of 0·5 lb. per B.H.P. hour. Thus a very substantial saving in fuel might be effected if some means were provided for reducing the quantity of the scavenging air on light loads.

Before proceeding to discuss the mechanical details it will be well to tabulate the results which have been arrived at at this stage. Later on, in this book, actual test figures will be quoted, and some of these are tabulated below with the figures arrived at from the foregoing reasoning. They will not be found to differ very widely. The typical engines that have been considered so far are supposed to represent the best modern practice. The two-cycle gas-engine has been purposely omitted at this stage, because its efficiency depends almost entirely upon stratification, which is far too uncertain a quantity to base any calculations upon. Taking in each case an economical load, generally about 85–90 per cent of the maximum load, we obtain the following figures:—

	Gas-engine	Four-cycle Diesel.	Two-cycle Diesel.
Bore inches	12		12
Stroke inches	18	18	18
R.P.M.	240	240	240
Piston speed, feet per minute	720	720	720
Brake horse-power (at economical load)	46 5	53 0	89·6
Indicated horse-power	53·5	70 5	123·4
Mechanical efficiency	87	75	72 %
Indicated thermal efficiency per cent	35	43	42
Compression ratio	6 : 1	14 : 1	14 : 1
Air-standard efficiency per cent	51·1	65	65 %
Maximum pressure, pounds per square inch absolute	496	518	518
Mean pressure, pounds per square inch	86·5	114	100
Brake thermal efficiency per cent	30·5	32·2 %	31·1 %

If the load be reduced to one-third, the results are as follows:—

Power controlled by
- (*a*) hit-and-miss.
- (*b*) qualitative governing.
- (b_1) qualitative governing and reduced scavenge air.
- (*c*) quantitative governing.

	Gas-engine.			Four-cycle Diesel.	Two-cycle Diesel.	
	(a)	(b)	(c)	(b)	(b)	(b_2)
Brake horse-power	15·5	15 5	15·5	17·6	30	30
Indicated horse-power ...	23·7	21·95	24·55	32·0	58·3	52·6
Mechanical efficiency per cent	65·7 %	70 6 %	63·2 %	55 %	51·5 %	57 %
Indicated thermal efficiency per cent	36 %	44 %	28 %	48 %	48 %	48 %
Brake thermal efficiency per cent	23·6 %	31 %	17·7 %	26·4 %	24·7 %	27·4 %

From the above figures it will be seen that, from the point of view of net thermal efficiency, there is but little to choose between the constant-volume and constant-pressure types, but, since quantitative governing is at present the only practicable method of controlling large explosion engines, the efficiency of the Diesel type is considerably higher on light loads.

The following results are collected from a series of actual tests:—

Maker's Name.	Gas-engines.			Diesel Engines, Four-cycle.		Diesel Engines, Two-cycle.	
	Swiss Locomotive Works.	Crossley.	National				
Number of cylinders	1	1	1	1	1	2	4
Bore, inches	10·6	11·5	14	23·6	12	15	23·6
Stroke, inches	16	21	22	33	18·25	22	33
R.P.M.	200	180	166	150	203	188	136
Piston speed, feet per minute ...	533	630	609	828	610	570	750
Compression ratio	—	6·37 : 1	5·34 : 1	14 : 1	14 : 1 about	14 : 1 about	14 · 1 about
Air standard efficiency per cent	—	52·2 %	49 %	65 %	65 %	65	65 %
Brake horse-power	22·9	38·7	52·7	800	39	200	1400
Indicated horse-power	27·9	14·5	61·3	1110	52	272	1940
Mechanical efficiency per cent ...	82	87	86	73	75	73·4 %	72
Indicated thermal efficiency ...	37·7	36	35	46·6 %	43·2 %	41·7 %	43·3 %
Brake thermal efficiency ..	30·9 %	31·3 %	29·9 %	34 %	32·4 %	30·5 %	31·2 %

The above results show clearly that, from the point of view of net thermal efficiency, the Diesel engine has on full load no pronounced advantage over the gas-engine. The indicated thermal efficiency is, however, considerably higher than that of the gas-engine. This result, however, is only to be expected so long as it remains necessary to devote so large a portion of the power output of the engine to the supply of compressed air. On the other hand, it has the in-

herent disadvantage that it may, under abnormal conditions, such as a leaky fuel valve or other cause, be subjected to very much higher pressures than the gas-engine. Suppose, for example, that owing to some mischance fuel entered the cylinder during the suction stroke, and was vaporized during the compression stroke to such an extent that the cylinder near the end of this stroke contained an explosive mixture of air and oil. This vapour would be ignited before the top of the stroke by the heat of compression, and the pressure which might conceivably be produced under those circumstances would be in the neighbourhood of 1500 lb. per square inch. That such a pressure as this would ever be produced in practice is wellnigh impossible, but a momentary rise of pressure up to and exceeding 1000 lb. per square inch is no uncommon occurrence in a Diesel oil-engine, and the whole structure of the engine must be made strong enough safely to withstand such pressure. It is a common claim that the Diesel engine is free from all danger of pre-ignition. There can be no greater fallacy.

The preceding calculations serve to demonstrate that the full-load indicated thermal efficiency of modern internal-combustion engines has approached so near to the ideal efficiency for each particular cycle, that no very great improvement can be hoped for, unless some entirely new cycle be introduced, or means be employed for utilizing the heat lost to the cooling water and the exhaust. Since further increase in the indicated thermal efficiency must necessarily be very limited, the attention of engineers and designers must be concentrated rather on the elimination of the mechanical and fluid losses. Particular attention should be given to the piston, which is the principal source of mechanical loss, and whose weight limits the speed of rotation, and therefore the power out-put from a given size of engine.

CHAPTER VIII

DETERMINATION OF THE MECHANICAL EFFICIENCY

In the preceding chapters it has been shown that the indicated thermal efficiency of a modern explosion engine of first-class design is generally within about 86–88 per cent of the ideal efficiency for the actual working fluid employed. It has also been shown that the mechanical efficiency is generally within about 86–88 per cent of the total indicated power. In other words, the scope for improvement both as regards the thermal and mechanical conditions is about 13 per cent in each case. The thermal conditions have been dealt with at considerable length, and it has been shown that of the 13 per cent losses, heat loss to the cylinder walls accounts for something like 10 per cent, and that any research in this direction should be devoted to eliminating the heat losses as far as possible. At the present time, and with the instruments available in most engineering workshops and test rooms, it is quite impossible to measure the mechanical efficiency of an internal-combustion engine with any approach to accuracy, because, although the brake horse-power can be determined within very fine limits, the indicated horse-power can only be ascertained by the use of the ordinary pencil indicator, which cannot be relied upon within about 5 per cent. Now, a 5-per-cent error in the indicated horse-power may easily result in a 40-per-cent error in the mechanical efficiency. With an optical indicator it is possible to ascertain the indicated horse-power to within about 1·5 per cent, but such an indicator is too delicate an instrument for the ordinary manufacturer's test room, and its use is practically restricted to experimental laboratories. Tests for mechanical efficiency are often made by driving the engine round by means of an electric motor and measuring the power absorbed, but such tests are of little real value, because both the friction and fluid losses are entirely different when the engine is not firing. Probably the most accurate method of measuring the

mechanical efficiency of a multi-cylinder engine is that devised by Mr. L. G. E. Morse, which consists in measuring the brake horse-power when all the cylinders are in normal operation, then cut out the ignition of one cylinder and again ascertain the brake horse-power of the remainder. If

n = number of cylinders,
B = total brake horse-power when all cylinders are firing,
b = total brake horse-power when $n - 1$ cylinders are firing,
I = indicated horse-power of each cylinder.

Then I = $n(B - b)$, which gives the indicated horse-power; and

$\frac{B}{I}$ = mechanical efficiency.

This method is, of course, only applicable to multi-cylindered engines, but it provides a reasonably accurate measurement of the mechanical losses, though it does not subdivide the losses or give any clue as to how they are incurred. It is needless to point out that it is very desirable to find some means of ascertaining the mechanical efficiency, for two reasons:—

1. Since only the brake horse-power can be measured with any reasonable degree of accuracy, the indicated horse-power and efficiency cannot be ascertained unless the mechanical efficiency is known.

2. It is hopeless to attempt to improve the mechanical efficiency unless some means be found for ascertaining where and how the losses are incurred.

Throughout this book the author has endeavoured to calculate the mechanical efficiency of the various types of engines described, and for this purpose he has made certain assumptions. These assumptions are based on the collective results of tests that have been published from time to time. The data available is very limited, and it will not be surprising if several of his assumptions require considerable modification when more light has been thrown on this important subject.

Of the 13-per-cent mechanical losses probably about 3 per cent are accounted for by fluid losses; that is to say, by work done on the gases in drawing them into the cylinder and expelling them. Strictly speaking, these are not mechanical losses, but they are always classified under that head, and since their magnitude is dependent upon the size and operation of the valves and other

essentially mechanical conditions, there is some justification for so classifying them. The loss due to bearing friction probably does not, as a rule, exceed about 3·0 per cent of the total indicated power. The remaining 7 per cent is apparently due to piston friction.

Dealing with these items one by one; provided there is no undue restriction in the exhaust or inlet pipes, the power absorbed in charging and discharging the cylinder depends upon the velocity of the gases through the valves, the larger the valves the lower the velocity and the smaller the loss from this source. In an explosion engine, however, it is desirable that the velocity of the entering gases shall be fairly high, in order that the charge within the cylinder shall be in a state of violent turbulence, a condition essential for rapid and complete combustion; for this reason very large inlet valves cannot be employed with advantage, for the saving in pumping loss is more than counteracted by the loss of efficiency due to incomplete combustion. In practice it is generally found that the best all-round results are obtained when the velocity through the inlet valves is between 100 and 130 ft. per second. These restrictions do not apply to the exhaust valve, but in this case a higher velocity can be safely employed, because the higher pressure of the exhaust gases at the moment of release creates so high a velocity in the exhaust pipe, that a certain amount of scavenging takes place owing to the inertia of the gases. Also, it is desirable to keep the diameter of the exhaust valve as small as possible, for three reasons:—

1. This valve has to be lifted against a comparatively high pressure, and since it is unbalanced, a severe strain is thrown upon the valve gear.

2. The valve is generally uncooled, and is heated by the exhaust gases which pass round it at an exceedingly high velocity. It can only get rid of the heat imparted to it by conduction down the stem or through the seating. If the diameter of the valve is large, distortion is liable to take place, and therefore leakage.

3. The valve, being uncooled, is liable to attain so high a temperature as to cause premature ignition.

These restrictions do not apply in the case of water-cooled exhaust valves, but such valves are so cumbrous and so liable to give trouble, due to leakage or failure of the water circulation, that they are never employed except when absolutely necessary.

In the case of Diesel engines a lower inlet velocity could no doubt be employed, for the necessary turbulence is produced largely

by the high velocity of the blast air entering with the fuel. But in these engines, owing to the necessity for distributing the fuel thoroughly and uniformly throughout the combustion chamber, it is essential that it shall be as compact as possible. This means that all the valves are in practice, congregated together in a flat cylinder head, and since there are four valves in all, including an air-starting valve, it is clear that none of them can be of large diameter. For this reason the gas velocities in Diesel engines are generally higher in proportion to the piston speed than in explosion engines.

If the same quantity of gas entered the cylinder at every cycle irrespective of the velocity through the valves, then it is clear that the work done upon it would be proportional to the square of the velocity, and that therefore the fluid losses would increase directly as the square of the speed. In practice, however, with a given area of valve opening the volumetric efficiency will decrease as the velocity is increased, because the pressure difference between the inside and the outside of the cylinder is relied upon to force the gases in. This pressure difference is comparatively small, therefore the fluid losses do not increase as the square of the speed, but as some function of it which has yet to be determined. Up to the present time very little investigation has been carried out, and there is hardly sufficient data available to determine what this function is; moreover, it is not dependent upon the effective area and velocity alone, but also upon what might be described as the nozzle coefficient of the orifice through the valve. For example, a circular or rectangular port will pass something like double the quantity of gas for the same effective area and pressure difference that an ordinary poppet valve will pass. Again, poppet valves which open directly into the main body of the combustion space will pass some 20 per cent more gas than those which open into a side pocket. The whole question of fluid loss during the pumping strokes requires thorough investigation, and such an investigation of this subject would be of great value to designers, since it would enable them not only to reduce the fluid losses which are already comparatively small, but also to increase the volumetric efficiency, which is of much more importance. The curve, fig. 22, illustrates the fluid losses in terms of the B.H.P. at varying speeds; the losses in this case were measured directly by means of an optical indicator while the engine was running under full load, and is one of the very few published tests in which this has been done. The engine used for these tests was a very small one, only

$3\frac{9}{16}$ in. bore by $5\frac{1}{8}$ in. stroke; and, unfortunately, the dimensions and lift of the valves are not recorded, so that no data can be deduced beyond the general shape of the curve; too much reliance

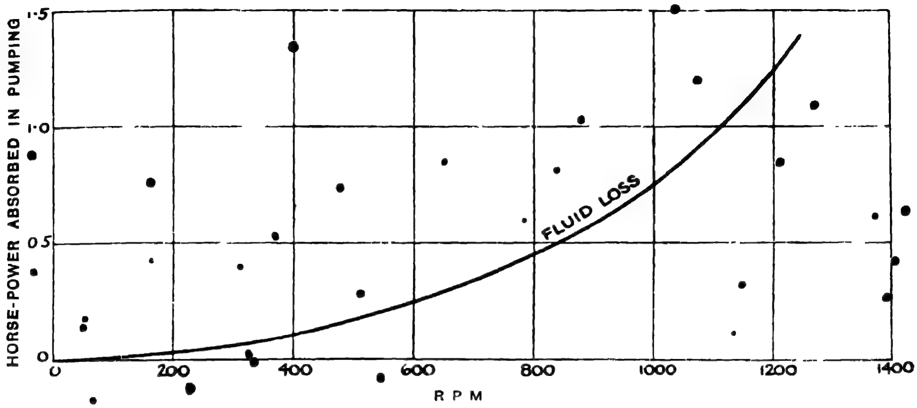


Fig. 22.—Curve showing Fluid Losses of Daimler Engine at various Speeds

must not, however, be placed upon this one example, which may not have been normal.

The curve shown in fig. 23 is computed from the volumetric efficiencies determined in a number of tests which have been carried

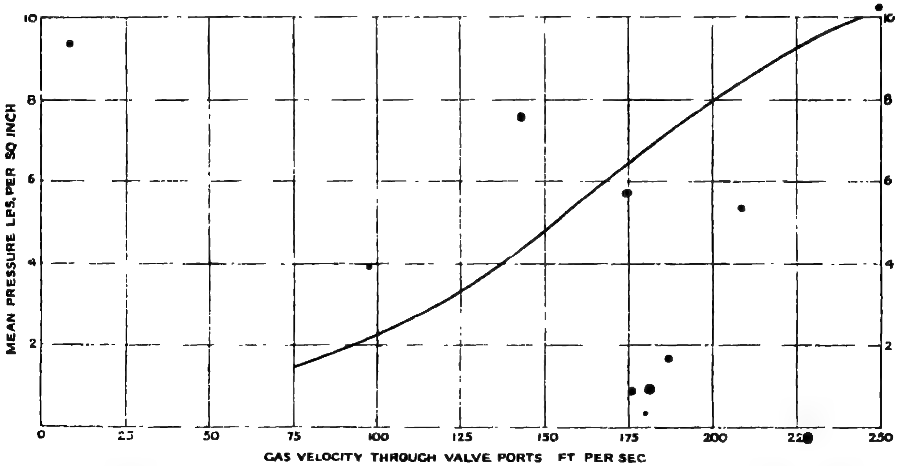


Fig 23.—Mean Pressure of Suction Loop for different Gas Velocities with Normal Valve Setting and Poppet Valves

out on various engines, mostly small engines of the high-speed type, by different experimenters. The horizontal readings are in terms of the velocity through the valves, and, as is usual, these are reckoned on the assumption that the volumetric efficiency is 100 per cent,

and that the valves are held fully open throughout the entire stroke. The vertical readings give the power absorbed in pumping in terms of pounds per square inch of piston area. The curve is obtained by calculating the actual velocity of the gases through the valve from the figures given for the volumetric efficiency, and assuming that the fluid loss is proportional to the square of the gas velocity. This method gives the fluid loss during the suction stroke only, the fluid loss during the exhaust stroke can only be guessed at; as a general rule, with equal-sized valves, the exhaust back pressure is about half the suction pressure, and this has been assumed to be the case in computing this curve. It has also been checked by comparison with a large number of light-spring indicator diagrams. It must be clearly understood that a curve such as this can only be regarded as approximate, and is offered only in lieu of more reliable data from actual measurement from a number of different engines; also, it is, of course, only applicable to normal conditions, and might be very wide of the mark if, for instance, there were any serious exhaust back pressure, or if the pipework were so arranged as to offer any serious resistance to the free passage of the inlet and exhaust gases.

Measurements of the fluid losses when an engine is being motored round are of little or no value, because:

1. The work done on the gases during compression is not returned in full during the expansion stroke; heat is given up to the cylinder walls and is not all returned.

2. The exhaust back pressure is very much greater, because there is no pressure in the cylinder when the exhaust valve is first opened, and consequently no kinetic energy is available to help withdraw the exhaust gases.

For both these reasons the fluid losses, as measured by motoring the engine round, are far too high. From the above considerations it would appear that, so long as the turbulence necessary for rapid combustion is produced by the high velocity of the entering gases, these fluid losses are not susceptible of any great improvement; but they can be somewhat reduced by careful design, both of the passages leading to the valve, of the valve opening itself, and of the walls of the cylinder surrounding the valve, with a view to improving the nozzle coefficient.

It is quite conceivable that the necessary turbulence might be produced during the compression stroke, as, for example, by forcing the air or gases into a bulb with a restricted neck, as is done in the

case of some of the semi-Diesel engines, but from the point of view of the fluid losses this is only transferring the negative work from one idle stroke to another. Again, it might be produced actually at the moment of ignition by allowing a portion of the gases to pass into a small pocket communicating with the cylinder through a narrow neck; if ignition be commenced in this pocket, the rapidly expanding and burning gases issuing from the neck will produce all the turbulence that is required in the main body of the combustion space. If this is done, very much larger inlet valves can be employed with advantage, and the fluid losses considerably reduced.

For the purpose of calculating the fluid losses, the curve, fig. 23, has been made use of throughout this book.

For convenience of calculation, the fluid and other losses have been expressed in most cases in terms of pounds per square inch of piston area in preference to percentage of total I.H.P., which would vary according to the mean effective pressure.

Bearing Friction.—Under this heading is included not only the friction of the main bearings, but also that of the camshaft and the power absorbed in operating the valves and driving such auxiliaries as magnetos, &c.

The bearing friction, as distinct from piston friction, cannot very well be ascertained while the engine is in operation. By the use of an accurate optical indicator it is possible to ascertain the total fluid and mechanical losses with a very fair degree of accuracy, and also to determine the fluid losses separately, so that the total mechanical losses can be arrived at, but it is not possible to subdivide these.

The only method of ascertaining the bearing as distinct from the piston friction is by motoring the engine round with the pistons removed; this gives too low a figure, for the following reasons:—

1. The removal of the pistons involves the removal of the connecting-rods also, and hence the bearing friction of the connecting-rods is not included in the total.

2. The removal of the pistons involves the removal of all load on the bearings due to the fluid pressures and the inertia of the reciprocating parts.

3. The exhaust valve is not opened against pressure.

In the case of single-cylinder slow-running engines the mean load on the bearings, due to the weight of the fly-wheel, is generally considerably in excess of the mean fluid and inertia pressures throughout the cycle, and therefore the removal of the pistons does not greatly affect the accuracy of this method of determining the

bearing friction. In high-speed multi-cylindered engines the fly-wheel weight is small in comparison to the mean fluid and inertia pressures, hence the degree of accuracy is much lower. For this reason more weight should be given to those tests which are carried out on single-cylinder engines. Also, owing to the use of a much heavier fly-wheel, the bearing friction is considerably greater in single-cylinder engines.

Professor Riedler has carried out some very interesting investigations in order to discover the power absorbed in operating the valves while the engine is running under normal full-load conditions. For this purpose the camshaft of the engine was driven

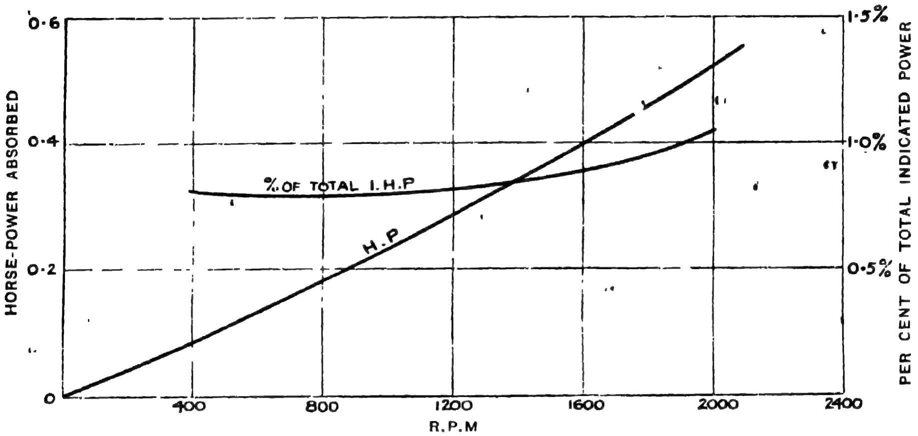


Fig. 24.—Power absorbed by Valve Gear at Varying Speeds—Adler Engine, 35 B.H.P.

independently by an electric motor, and the driving chain connecting the crankshaft and camshaft was employed only to ensure the correct relative motion of the two, but not to transmit any power. This was accomplished by so adjusting the power of the motor that the driving chain was slack on both sides. The investigations were carried out on a small four-cylindered petrol engine running under full load and at various speeds; the valve gear of such an engine will absorb more power than in the case of most other types of internal-combustion engine, because owing to the high speed, the valves are relatively much larger and the valve springs much stiffer than usual. The results of the investigations are given in the curve (fig. 24), from which it will be seen that the power absorbed by the valve gear varies almost directly as the speed. In this case, however, it appears that the power absorbed in driving the auxiliaries, consisting probably of a magneto and oil-circulating pump, was included

in that of the valve gear, for it appears from a photograph of the engine tested that these auxiliaries were driven from the camshaft.

The curve (fig. 25) shows the power absorbed in bearing friction and valve operation of a six-cylinder Pierce-Arrow engine, in terms of pounds per square inch of piston area, as ascertained by Mr. Herbert Chase at the New York Automobile Club's Laboratory. In this case the pistons and connecting-rods were removed, and the engine motored round by an electric motor at various speeds. This engine had a very small fly-wheel, so that the results obtained will be very considerably below the true figure when the pistons are replaced, and the engine running under normal full-load conditions. On the other hand, the bearing friction in this case

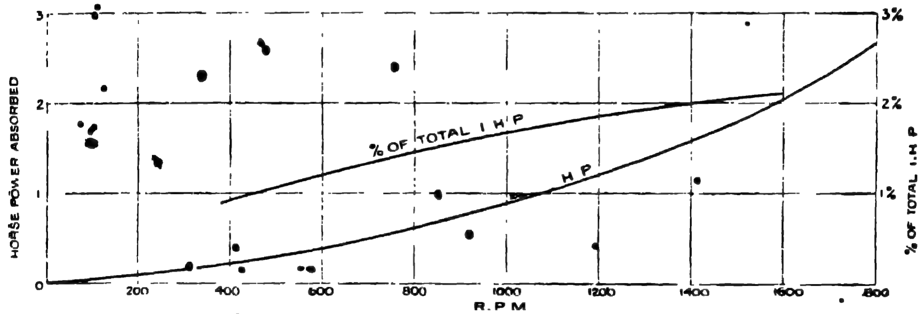


Fig. 25. Power absorbed by Bearing Friction and Valve Operation—Pierce-Arrow Engine, 80 B.H.P.

includes the power absorbed in driving, not only the magneto, but also the oil and water circulating pumps.

Professor Hopkinson, in his tests in the laboratory of Cambridge University on a Crossley gas-engine having one cylinder 11·5-in. bore and 21-in. stroke, with a normal speed of 180 R.P.M., found that the loss due to bearing friction and valve operation amounted to 2·7 per cent of the total indicated horse-power. This engine had two heavy fly-wheels, and relied upon ring lubrication of the bearings, so that the bearing friction would be above the average in this case.

Mr. Pomeroy, in his tests on a four-cylinder Vauxhall racing petrol engine of 4 bore and 5·5 stroke, found that the total power required to overcome bearing friction and operate the valves when the pistons and connecting-rods were removed amounted, at a speed of 3700 R.P.M., to approximately 4 horse-power, equal to about 3 per cent of the total I.H.P. at this speed, or to about 3 lb. per square inch on the piston. This test is of particular interest, because the speed of rotation is exceedingly high, yet the bearing friction has not risen to any serious extent above that

obtained in tests of similar engines at comparatively low speeds. In this engine the power required to operate the valves would be abnormally high, because not only were exceedingly heavy springs employed, but each cylinder was provided with two inlet and two exhaust valves, so that the power absorbed by the valve gear would be nearly double that in Professor Riedler's tests.

With the exception of Professor Riedler's investigations into the power absorbed by the valve gear, all the above tests were carried out with the pistons and connecting-rods removed, and therefore the readings obtained are all below the true figure, how much below it is not possible to say with any degree of accuracy. Professor Hopkinson, however, has stated that he has found that the friction loss in the case of the single-cylinder engine is practically constant under all conditions of load. It is fairly clear that under these conditions of testing the operation of the normal valve gear amounts to about 33 per cent of the total losses under this heading, and that the actual bearing friction alone does not exceed about 66 per cent. The actual figure depends both upon the type of bearings, system of lubrication, and weight of the fly-wheels.

These figures serve to show that the loss due to bearing friction is in any case extremely small, and that it is not worth while devoting much attention to its further reduction. It is of far more importance to ensure that the bearing surfaces are ample and efficiently lubricated. It is clear that the advantage to be gained by the substitution of ball bearings is a trifling one from the point of view of mechanical efficiency; their real advantage lies in the fact that they require very little lubrication, and in small engines at all events, where the loading is often severe and the area restricted, ball bearings are considerably more reliable, and will run for much longer periods without requiring adjustment or renewal. It is also fairly clear that the friction under this heading does increase with increase of speed, but the increment is small, and since the total friction losses included under this head are very small, the increase may for convenience of calculation be neglected. It will, therefore, be approximately correct to assume that in all normal engines the proportion of the total I.H.P. absorbed in overcoming bearing friction and operating the valves and auxiliaries amounts to between 2 and 3 per cent of the total I.H.P., and since in each of the tests quoted the normal mean pressure of the engine is approximately 100 lb. per square inch, it follows that the losses are equal to a mean pressure on the piston during the power stroke of between 2 and 3 lb. per square inch.

CHAPTER IX

PISTON FRICTION

Piston Friction.—As a general rule, piston friction constitutes by far the largest item in the list of mechanical losses.

The immediate and direct cause of the piston friction is probably the shearing of the oil film formed between the cylinder and piston, and the power absorbed is dependent upon the thickness of the film, the viscosity of the oil, the area, and the speed.

The coefficient of friction between the cylinder and piston is very much higher than that of the bearings, and this probably is to be accounted for:—

1. By the reduced supply of lubricant as compared with the bearings.
2. The thickening of the lubricant due to partial carbonization, which greatly increases its resistance to shear.

1. If the supply of lubricating oil to the piston be profuse, then a considerable proportion of this oil will find its way into the combustion space, where it will produce pre-ignition, both directly and indirectly. Directly, by coming into contact with some highly-heated part, such as the exhaust valve, and being "cracked" or broken down into lighter and chemically unstable fractions which will ignite at a lower temperature than the working fluid; and indirectly, by forming a deposit of carbon on the walls of the combustion chamber and piston, which, being a poor conductor of heat, reaches a sufficiently high temperature to set up pre-ignition.

2. Owing to the high temperature attained by the head of the piston, a certain proportion of the lubricant is bound to be partially carbonized, and so thickened. In engines employing liquid fuel, unless combustion be absolutely complete and precipitation of the fuel on the walls of the cylinder completely eliminated, a portion of

the partially carbonized fuel oil will mix with the lubricant, and so thicken it and impair its lubricating properties. Further, the film adhering to the walls of the cylinder barrel is, at every cycle, exposed to the full flame temperature.

In order to prevent carbonization as far as possible an oil must be employed which will withstand a high temperature without decomposition; such an oil has a very high viscosity and therefore a high resistance to shear.

From such experiments as have been made on piston friction, it appears that this varies nearly as the square of the rotative speed and directly as the piston speed; that is to say, the piston friction of an engine with a 20-in. stroke running at 200 R.P.M. would be doubled if the stroke were increased to 40 in. with the same rotative speed, but if the rotative speed were increased to 400 R.P.M. and a 20-in. stroke retained, then the piston friction would be nearly four times as great.

Now, if the thickness of the oil film were constant, the resistance to shear would presumably be proportional to the piston speed, whether this were varied by increasing the rotative speed or the piston stroke.

The thickness of the oil film is, however, dependent upon the pressure between the surface of the piston and that of the cylinder walls. The pressure of the piston against the cylinder for a given ratio of connecting-rod to crank $\frac{l}{r}$ is dependent upon the fluid pressures in the cylinder, and upon the inertia pressure. The fluid pressures remain approximately constant over a wide range of speed, but the inertia pressures vary as the square of the rotative speed, and directly as the piston speed. Now, as a general rule, most four-cycle internal-combustion engines run at such a speed that the inertia pressures exceed the fluid pressures when both are averaged over the whole cycle, and therefore piston friction depends mainly upon the inertia pressures, and increases as they increase; that is to say, other things being equal, the piston friction is mainly dependent upon the weight of the reciprocating parts.

Clearance of Piston.—It is very desirable to keep the clearance between what may be termed the cross-head portion of the piston and the cylinder walls down to the lowest possible limit consistent with leaving provision for expansion and distortion. If an unduly large clearance is permitted—

1. The noise set up due to the piston crossing violently from one side of the cylinder wall to the other may be very serious.

2. Any tilting or other movement of the piston is objectionable, in that it causes the rings to slide in their grooves, and so induces wear.

3. When the ring or ring-grooves become worn there is danger of the burning gases passing down between the cross-head portion of the piston and the cylinder walls at a rate sufficient to scour off most of the lubricating oil and carbonize the remainder; thus causing greatly increased piston friction, and even actual seizure under extreme conditions.

A very small quantity of gas leaking past the piston rings, if allowed to pass down between the trunk and the cylinder walls, will very soon carbonize the oil to a degree sufficient greatly to increase the friction, even though the quantity of gas lost by leakage may be insignificant from the point of view of power or efficiency. It would seem desirable always to provide a free vent for any gases which pass the piston rings; this can be done very easily by turning a groove round the outside of the piston immediately below the bottom ring and allowing communication with the atmosphere by drilling a number of holes in this groove through the walls of the piston. When this is done the leakage of gas becomes more audible even when the rings are fitting well, though it is doubtful whether the actual quantity which escapes is any greater. Experience has shown in many cases that the reduction both in friction and in the quantity of oil working past the rings into the combustion chamber is very marked, and the resulting gain in power and economy certainly more than outweighs any possible increase in leakage.

All these considerations presuppose that both the piston and cylinder are truly circular, and remain so when heated. In practice this is not the case, and a certain amount of distortion always occurs. In those engines in which a separate liner is employed, which is simply a plain cylinder of uniform thickness, the amount of distortion is probably inconsiderable. Distortion of the piston is generally due to the gudgeon pin bosses, which locally spoil the symmetry of this part, and also it is in part due to the gudgeon pin, which is necessarily a tight fit in the bosses. The evil effects of such distortion can be largely avoided by cutting away or relieving the sides of the piston in the neighbourhood of the gudgeon pin, so that any distortion which may occur will not cause the piston to

bind in the cylinder. This does not reduce the available bearing surface, because the material is only cut away at the sides, and not on the working faces.

It appears evident that slackness of the rings in their grooves is one of the most fruitful causes of the passage of oil into the combustion chamber. On the outward stroke of the piston the rings will, owing in part to friction against the cylinder walls, and in part to inertia, bear closely against the upper faces of their grooves, leaving a gap below their lower faces which will be partly filled with oil. As the piston changes its direction at the end of the outward stroke, the rings will change over and bear against the lower faces of their grooves, and a large proportion of the oil expelled by this movement will pass behind the rings into the upper space. In this manner at every reversal of the piston a substantial proportion of the oil in the ring-grooves will be pumped up past the rings, and so progressively into the combustion chamber. However closely the rings may be fitted, this action takes place to some extent, but the quantity of oil pumped depends very largely upon the side clearance of the rings, and is almost independent of the number employed. It would appear that if free vents were provided at the back of one of the rings by drilling holes through the bottom of the ring-groove, open to atmosphere or to the crankcase, a considerable proportion of the oil would escape through these vents instead of passing up into the combustion chamber. So far as the author is aware this method of checking the passage of oil into the combustion chamber has not yet been tried.

To sum up, it seems fairly evident that to reduce piston friction to the lowest possible limit the clearance should be kept as small as is consistent with the distortion that may take place. Unless the supply of lubricant be copious, and means be provided for permitting of the free escape of any gases that may pass the piston rings, the evils and dangers attendant upon considerable clearance greatly outweigh the advantages, for what is gained by the use of a thicker oil film is more than outweighed by the greater consumption of oil, the greater resistance due to carbonization of the lubricant on the piston walls, the more rapid carbonization of the combustion chamber, and the risk of scoring and seizing.

The above remarks refer only to the clearance between what may be described as the crosshead portion of the piston and the cylinder walls. That portion above and between the piston-rings may be allowed much greater clearance, because its only function is

to maintain the piston rings in place; and since it is very much hotter greater expansion and distortion take place. The clearance allowed in this part should be sufficient to ensure that it does not come in contact with the cylinder walls under any conditions, for the temperature of the metal is so high that it is not possible to lubricate it. It is generally found that whatever clearance is permitted above the rings is rapidly filled up with a deposit of hard carbon—this carbon, however, shows little or no tendency to score the cylinder walls. Since there is always a gap between the ends of the piston rings it follows that the greater the clearance between the piston and cylinder walls the more is this gap exposed and the greater the leakage. It is therefore desirable to allow only just as much clearance as is necessary to ensure that this part of the piston shall not come into actual contact with the cylinder walls when heated to the highest working temperature; for this reason it is preferable to machine the diameter of this part in steps providing the maximum clearance above the first ring where the temperature is at a maximum and reducing it progressively between the rings.

Effect of Temperature.—It is almost invariably found that the piston friction is reduced in a very marked degree as the temperature of the cylinder walls increases. If, as may be supposed, the friction in any given engine is dependent upon the viscosity of the oil, then it follows that this will be reduced as the temperature rises and the viscosity decreases. The curves (fig. 26) show the difference in the mechanical efficiency of a small four-cylinder Daimler engine with different jacket temperatures. Since neither the fluid nor the bearing friction can be appreciably affected by comparatively slight changes in the wall temperature, it follows that the higher mechanical efficiency at the higher jacket temperatures is due solely to reduction of piston friction.

Professor Hopkinson in his tests on the Crossley engine, already alluded to, found that the piston friction varied from 10 per cent of the I.H.P. when the jacket temperature was 70° F. to 6·1 per cent when it was raised to 180°.

Coefficient of Friction.—When the actual pressure of the piston on the cylinder walls due to the angular thrust of the connecting-rod is considered, it will be found to amount to only a very low figure, and it is somewhat surprising that the coefficient of friction should be so high. The explanation appears to be that the lubrication of the ordinary trunk piston is more or less defective;

this is probably due to the carbonization of the lubricant. Such carbonization is no doubt caused both by the passage of a small quantity of highly heated gases between the cylinder and piston walls, and by the high temperatures of the upper parts of the piston. In the case of oil-engines, and especially those in which combustion is not very complete, the lubricant is largely adulterated with partially-burnt fuel oil, and the coefficient of friction increased accordingly. If the cross-head portion of the piston be made separate from the piston head and kept cool the friction can be reduced enormously, for in

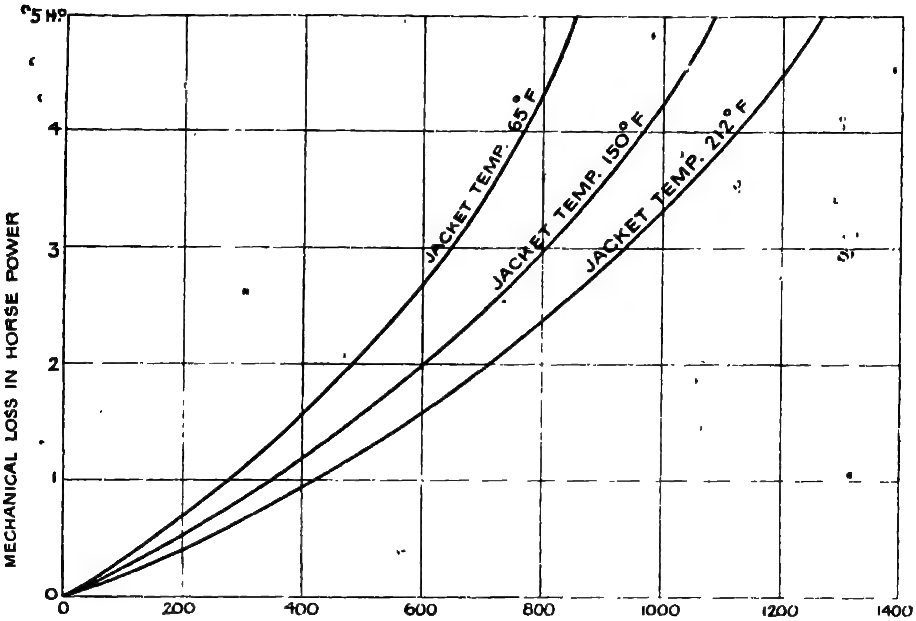


Fig. 26.—Curve showing Mechanical Loss of Daimler Engine at Various Speeds and Jacket Temperatures

that case this portion can be profusely lubricated without any fear of the oil becoming carbonized, and since the lubricant is not subjected to high temperatures a very much thinner oil can be safely employed. Experience with double-acting engines employing an external cross-head has shown that the piston and cross-head friction combined is generally only about half that of an ordinary trunk piston.

As proving that under normal conditions the lubrication of trunk pistons is defective, Professor Hopkinson and others have carried out experiments showing that if the pistons be profusely lubricated with a mixture of oil and water, or if the lubricant be thinned by any other means, the piston friction can be reduced to little more than

one-half of the normal, but these conditions persisted only so long as a copious supply of lubricant was employed, and could not be maintained under practical conditions of working. The actual figures obtained were as follows:—

Normal piston friction	6·1 per cent of I.H.P.	
With excessive lubrication with oil and water	3·1	about.

This experiment can be demonstrated in the case of any gas-engine controlled by hit-and-miss governing. In such an engine the mechanical efficiency can be gauged to some extent by noting the proportion of explosions to misses when running light. Under normal conditions the proportion generally is about 5 or 6 missfires to each explosion; if, now, an emulsion of oil and water be admitted through the air inlet valve, it will be noticed that the proportion of missfires will increase to about 7 or 8 : 1, showing that the mechanical efficiency has been improved, and this will continue so long as the supply of emulsion is fed into the cylinder. From the very few published experiments in which any attempt has been made to ascertain the coefficient of piston friction of an engine, it appears that under normal conditions as to clearance, lubrication, and temperature, and with a ratio of connecting-rod length $\frac{l}{r} = 5$, the coefficient of friction amounts to approximately 3 per cent of the total pressure on the head of the piston, referred to the power stroke only. These experiments, however, have all been carried out on either gas or petrol engines. In engines using oil fuel there is evidence that, owing to contamination of the lubricant by partially-burnt fuel, the coefficient of friction is considerably higher than this.

Constant Friction.—The friction of the cross-head portion of the piston varies very nearly as the square of the speed and a large number of other conditions, but there is a certain amount of constant friction due to the pressure of the piston rings against the cylinder walls. This source of friction is dependent upon the stiffness of the piston rings, and so long as the rings are free in their grooves it is apparently nearly independent of the speed or any other conditions. There is a good deal of misapprehension as to the proportion of piston friction due to the piston rings. As a general rule, it does not amount to more than about 0·3 lb. per square inch of piston area, and is often even less; provided, of course, that the engine has run for

a long enough period for the working faces of the rings to be thoroughly bedded in and polished. The approximate value can easily be ascertained in the case of small pistons, by placing the piston with its rings in the cylinder, and noting at what angle of inclination to the horizontal it will slide down the cylinder barrel. If the weight of the piston and the angle of inclination are ascertained, the pressure per square inch necessary to overcome the constant friction can be arrived at. In oil-engines both of the vaporizing and semi-Diesel type, there appears to be a considerable amount of "gumming" of the piston rings due to partially-burnt fuel, and there is some reason for supposing that this greatly increases the friction of the piston rings, and therefore the constant friction due to them.

Area of Bearing Surface.—The area of the bearing surface of the piston has a considerable influence on the piston friction, and here again opinions differ very greatly. It is generally argued that the greater the surface the less the load per square inch, and, therefore, the less the wear of both the piston and cylinder walls. This certainly appears an obvious argument, but it is nevertheless open to question whether it is altogether a sound one. To begin with, the greatest average pressure, except in unusually slow-running engines, is that due to the inertia of the reciprocating parts. Now, any increase in the bearing surface beyond a certain limit involves a corresponding increase in the weight of the piston, for although it may be argued that the increased bearing surface can be obtained by merely increasing the length of the skirt of the piston, yet for this extra surface to be of any value the skirt must be stiffened to obtain absolute rigidity to such an extent that the increase of weight is almost proportional to the increase of surface. If this be the case, and considering the inertia forces only, it is clear that the sole effect of increasing the surface has been to increase the total pressure, the pressure per square inch remaining the same. The wear on the piston will be the same, but that on the cylinder walls will be greater, for the wear of the latter is dependent upon the total pressure. Hence, considering the inertia forces alone, the net result of increasing the bearing surface by lengthening the piston will be to increase both the piston friction and the wear of the cylinder.

When the fluid and inertia pressures are reckoned out in their true proportion, it will often be found that any increase in bearing surface obtained by lengthening the piston will result in a serious increase of the piston friction due to the greater area of the oil film,

no very material reduction in the pressure per square inch of bearing surface, and a considerable increase in the wear of the cylinder walls. That is to say, all that has been accomplished by so increasing the bearing surface has been to transfer the wear from the piston, which is comparatively cheap and easily renewed, to the cylinder, which is more expensive and much less easily renewed, and a considerable increase in the piston friction brought about. The whole question of bearing surface from the point of view of wear is intimately bound up with the proportion which the inertia pressures bear to the fluid pressures.

Again, taking the other extreme, it is clear that beyond a certain limit in the other direction any further reduction in the bearing surface will effect only a very trifling reduction in the weight of the piston, and thus the bearing pressure per square inch will be greatly increased, with the result that the wear of the piston will be rapid and the reduction in friction small, for what is gained by reducing the area is lost by the reduction in the thickness, and hence the greater resistance of the oil film.

It would appear that so long as the clearance is cut down to the lowest possible limit, in order to ensure the maintenance of an oil film all round the piston, the bearing surface may safely be reduced far below present-day practice.

Experience with high-speed petrol engines has shown that, from the point of view of piston friction, it is advantageous to reduce the bearing surface to the lowest possible limit, and it has also shown that the wear both on the cylinder walls and pistons is surprisingly small.

In the case of racing petrol engines it has been found that the piston friction can be still further reduced by drilling holes in the wall of the pistons, and thus not only still further reducing the bearing surface, but also allowing some of the oil entrapped between the cylinder walls and piston to escape, and so reduce the fluid resistance of the lubricant; but it must be remembered that in these engines the pistons receive far more lubricant than in other types, because the crank chambers are always totally enclosed, and, owing to the high speed and profuse lubrication of the bearings, oil is splashed on to the cylinder walls in great quantity.

Inertia of Reciprocating Parts.—It has already been pointed out that the pressure of the piston on the cylinder walls, due to the inertia of the reciprocating parts, generally exceeds the

average fluid pressure in the cylinder, and that this therefore is the main factor producing piston friction. The maximum inertia pressures are, of course, far below the maximum fluid pressures, except in the case of exceptionally high-speed engines. From the point of view of piston friction, however, it is not the momentary maximum pressures that need be considered, but the mean average pressure throughout the whole cycle.

The inertia pressures at either end of the stroke, in terms of pounds per square inch on the piston head, may be calculated from the formula:

At the commencement of the stroke $F_1 = \dfrac{w \times v^2}{g \times r}\left(1 + \dfrac{r}{l}\right);$

At the end of the stroke $F_2 = \dfrac{w \times v^2}{g \times r}\left(1 - \dfrac{r}{l}\right).$

w = weight of reciprocating parts, in terms of pounds per square inch of piston area,
r = radius of crank in feet,
v = velocity of crank-pin in feet per second,
g = gravity = 32·2,
l = length of connecting-rod in feet,
F = pressure on piston due to inertia, in terms of pounds per square inch.

It is interesting to consider the case of an engine of, say, 16-in. bore by 24-in. stroke, running at a speed of 200 R.P.M., and in which

$$w = 5 \text{ lb. per square inch},$$
$$r = 1,$$
$$v = \frac{2\pi \times 200}{60} = 21 \text{ ft. per second},$$
$$g = 32{\cdot}2,$$
$$l = 5 \text{ ft.}$$

In this case $F_1 = \dfrac{5 \times 21 \times 21}{32{\cdot}2}\left(1 + \tfrac{1}{5}\right)$

$= 82{\cdot}2$ lb. per square inch.

$F_2 = \dfrac{5 \times 21 \times 21}{32{\cdot}2}\left(1 - \tfrac{1}{5}\right)$

$= 55$ lb. per square inch.

These are the maximum inertia pressures, in terms of pounds per square inch of piston area at either end of the stroke. It will be

noticed that the inertia pressure at one end is considerably greater than at the other, due to the finite length of the connecting-rod. Since, however, from the point of view of piston friction, the maximum pressure is of no account, the influence of the length of the connecting-rod need not be considered in this respect, and the following very much simpler formula can be employed:—

$$F = 0{\cdot}00017\, w\, n^2 s \times \tfrac{1}{2},$$

where w = weight of reciprocating parts, in pounds per square inch,

n = revolutions per minute,

s = length of stroke in feet.

This formula gives the mean average inertia pressure during one stroke, and in order to place the inertia pressure on a par with the fluid pressure as usually reckoned it must, in a four-cycle engine, be multiplied by four.

In this case the mean inertia pressure during any one stroke becomes

$$F = 0{\cdot}00017 \times 5 \times 200 \times 200 \times 2 \times \tfrac{1}{2}$$
$$= 34 \text{ lb. per square inch,}$$

or $34 \times 4 = 136$ lb. per square inch when referred to the power stroke only. If the coefficient of piston friction be taken as 3 per cent, then the piston friction due to the inertia forces alone, and expressed in terms of pounds per square inch of piston area, will amount in this case to

$$\frac{3}{100} \times 136 = 4{\cdot}08 \text{ lb. per square inch.}$$

As a further example, it is interesting to consider the inertia pressures in the case of a high-speed petrol engine having, say, a 6-in. stroke, and running at a speed of 3000 R.P.M., a not uncommon speed for a modern high-speed engine. In this case the weight of the reciprocating parts will be reduced to the lowest possible limit, and will probably not exceed 0·3 lb. per square inch of piston area. The mean inertia pressure referred to the power stroke becomes

$$F = 0{\cdot}00017 \times 0{\cdot}3 \times 3000 \times 3000 \times \tfrac{1}{2} \times \tfrac{1}{2} \times 4$$
$$= 459 \text{ lb. per square inch,}$$

or vastly in excess of the fluid pressure. If the coefficient of piston

friction be again taken as 3 per cent, then the power absorbed in overcoming piston friction is equal to

$$\frac{}{100} \times 460 = 13{\cdot}8 \text{ lb. per square inch of piston area,}$$

or about 12 per cent of total indicated horse-power is absorbed in overcoming the piston friction due to the inertia forces alone. The second example is sufficient to indicate how very large a proportion of the total loss may be due to piston friction produced by the weight of the reciprocating parts, and how vitally important it is that the weight of these shall be reduced to the lowest possible limit.

The minimum weight of the reciprocating parts of an engine is governed by the maximum pressure which they have to withstand under the most extreme abnormal conditions, that is, in the event of premature ignition. In the case of explosion engines the pressure may under these circumstances rise to as high as $4\frac{1}{2}$ times the compression pressure. In Diesel or semi-Diesel engines it is hardly conceivable that the maximum pressure could exceed about $3\frac{1}{2}$ times the compression. Now, since the mechanical efficiency of an engine depends very largely upon the weight of the reciprocating parts, it follows that the engine which can show the lowest ratio of abnormal maximum pressure to mean pressure will have the highest mechanical efficiency. The mean pressure must be considered as being spread over the whole cycle. The following figures show the effect of this ratio upon the mechanical efficiency of a number of different types of internal-combustion engines. The figures are averages, and in each case the piston speed is assumed to be 750 ft. per minute:—

Type.	Compression Pressure.	Abnormal Maximum Pressure.	Mean Positive Fluid Pressure over Whole Cycle.	Ratio Maximum to Mean Pressure.	Mechanical Efficiency.
Petrol ...	99	405	27	15 : 1	90 %
Gas-engine ...	150	675	23	29·1 : 1	86 %
Semi-Diesel	250	875	20	43·8 : 1	82 %
Diesel ...	450	1500	27	55·5 : 1	82 %[1]

In spite of the greater ratio of maximum to mean pressure the

[1] Exclusive of high-pressure compressor.

Diesel engine, exclusive of the high-pressure blast air compressor, shows as high a mechanical efficiency as the semi-Diesel; this is probably to be accounted for by the more complete combustion in the true Diesel engine, which results in better lubrication of the piston; also the Diesel engine requires, and generally receives, better workmanship and more careful treatment.

Fluid Pressure.—In most internal-combustion engines the mean positive fluid pressures amount to from 70 to 110 lb. per square inch; but the total fluid pressures, including the negative pressure during the compression and pumping strokes, are considerably greater than this. The mean positive pressure is the total pressure during the working stroke minus the compression pressure. The total pressure, however, is the mean pressure during the compression stroke plus the total pressure during the working stroke plus the pressure during the pumping strokes; that is to say, it is equal to the mean positive pressure plus twice the compression pressure plus the pressure during the pumping strokes. In an engine having a compression pressure of, say, 150 lb. per square inch the mean pressure of compression will be about 26 lb. per square inch; this must be doubled and added to the mean positive pressure; the pressure during the pumping strokes is generally so small that it may be neglected.

In considering the piston friction it is the total fluid and not the mean positive fluid pressure that must be taken into account. Fig. 27 shows a typical gas-engine indicator diagram having a mean effective pressure of 91 lb. per square inch, as it might be produced by a continuous indicator, from which it will be seen that the total fluid pressure, both positive and negative referred to the working stroke, amounts to 143 lb. per square inch.

Fig. 27*a* shows the total fluid pressure acting on the piston throughout the whole cycle.

Fig. 27*b* shows the inertia pressure.

Fig. 27*c*, the combined fluid and inertia pressure.

Fig. 27*d* shows the total fluid pressure as referred to the working stroke alone.

Fig. 27*e* shows the total inertia pressure in the same way; and

Fig. 27*f*, the combined fluid and inertia pressure.

It will be noticed that the fluid and inertia pressures are not always cumulative, but that at certain periods during the cycle they counteract one another, such, for example, as during the latter

part of the compression stroke, so that it is not correct to deduce them separately and then add them together.

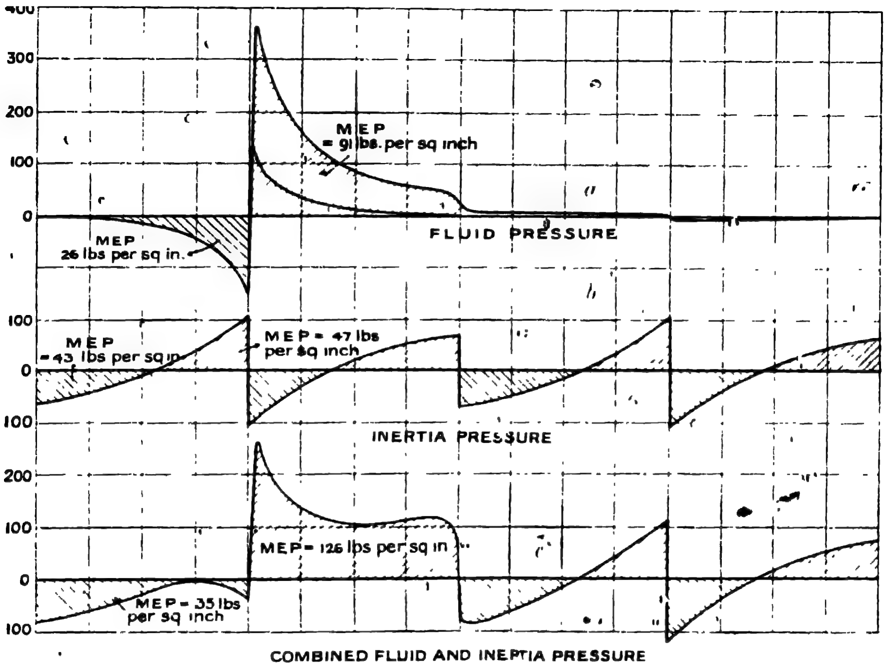
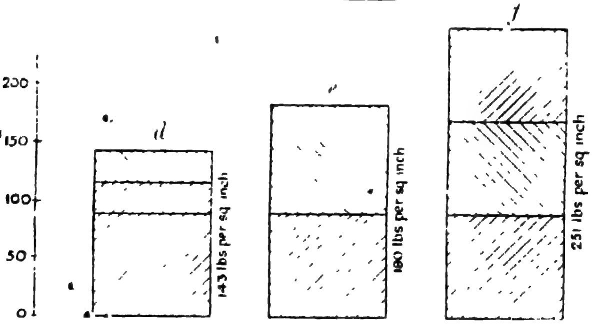


Diagram Showing Fluid and Inertia Pressure on a Four-cycle, Single-acting Gas-Engine Stroke 2 ft., R.P.M. 250, weight of reciprocating parts 4 lbs. per sq in.

Work done by crank on piston   Work done by piston on crank



Total fluid pressure referred to one stroke   Total inertia pressure referred to one stroke   Total combined fluid and inertia pressure referred to one stroke

Fig. 27

Influence of the Length of the Connecting-rod.—The length of the connecting-rod probably has some influence upon the piston friction, for it is obvious that the shorter the rod in proportion to the crank throw the greater is the angular thrust on the

cylinder walls and the greater the friction. Broadly speaking, the longer the connecting-rod the better, but it must be remembered that a longer rod means also a stiffer, and therefore a heavier one; since a certain proportion of the connecting-rod varying from one-quarter to one-half must be regarded as reciprocating weight, it follows that it would not pay to increase the length to an indefinite extent. In practice it is usual to make the length of the connecting-rod equal to five times the crank throw; that is, $\frac{l}{r} = 5$, and this applies to practically all internal-combustion engines with the exception of petrol engines, in which the ratio $\frac{l}{r}$ is sometimes as low as 3·7. The length of the connecting-rod also influences the side thrust of the piston, as upon it and the crank throw depends the obliquity of the thrust between the crank pin and the piston.

Formula for Deducing the Mechanical Efficiency.—It has already been explained that the indicated horse-power of an engine cannot be arrived at with any high degree of accuracy with the ordinary pencil indicator, on account of the inertia of the mechanism, the smallness of the diagram, and the unfavourable ratio between the maximum and mean pressures. Professor Hopkinson considers that even under the best conditions the pencil indicator cannot be relied upon to record the indicated horse-power within 5 per cent. Many of the indicator diagrams published in this book will, if integrated, show mechanical efficiencies varying in one instance from 74 per cent to 93 per cent for the same engine, and in another instance the indicated horse-power as recorded by the indicator is actually less than the brake horse-power. With the optical indicator far more accurate results can be obtained, because:

1. The pencil and multiplying levers are replaced by a ray of light, which has no weight, and therefore no inertia.

2. Any degree of multiplication can be obtained at will, and thus a larger diagram can be given with a smaller movement of the piston or diaphragm.

3. The multiplying mechanism consisting only of a ray of light is frictionless, and hence a very much smaller and lighter piston can be used without errors due to friction.

The optical indicator, however, is necessarily a very delicate instrument, and somewhat too fragile for use in an ordinary test room.

It is needless to point out that it is very important to ascertain the indicated horse-power of an engine, for unless this is known the designer is working in the dark as to the thermal conditions that obtain in his engine, and he cannot distinguish between the thermal and mechanical losses.

Throughout this book the author has endeavoured to calculate the fluid and mechanical losses by means of a formula. In explaining this formula, it is necessary to say at the outset that it is purely empirical, and, at best, merely an approximation, but it is probably considerably more accurate than the ordinary indicator. For a formula of this sort to be of any practical value it must be a simple one, and therefore it can only take into account a few of the more important variables. It will not be at all surprising if, when more is known as to the causes of the mechanical losses, it will be necessary to modify it very considerably, but in the present state of knowledge it gives tolerably correct readings when applied to the few cases in which accurate tests of mechanical efficiency have been made, and this is really its only justification.

The formula presupposes several conditions which have not yet been satisfactorily established. For example, it assumes that the piston friction is a definite percentage of the pressure on the cylinder walls. Also, it assumes that the friction of the bearings and valve-operating mechanism is the same under all conditions of speed and load. This latter assumption is admittedly incorrect, but the proportion of bearing friction is so small, that it is not worth while to complicate the formula for the sake of correcting the small error introduced. The following are the assumptions made, but in the practical application of the formula they should, of course, be varied at discretion to suit the special conditions in each case.

1. That the fluid losses, that is to say the area of the suction loop during the pumping strokes, is as given by the curve fig. 22, and is dependent solely upon the velocity through the valves. It does not take into account the size of the valves, which has a certain influence upon the nozzle coefficient. It assumes that the valves open into side pockets, as is generally the case, but that the walls of the cylinder surrounding the pockets admit of free entry and do not restrict the effective valve area. For valves opening directly into the cylinder head the proportion of fluid loss should be reduced by some 20 25 per cent. With a mean gas velocity through the valves of 130 ft. per second, the fluid loss is considered as equal

to a mean effective pressure of 3·5 lb. per square inch of piston area.

2. That the bearing friction is a constant depending to some extent upon the weight of the fly-wheel and the number and nature of the auxiliaries included under this head. The minimum figure is taken as 2 lb. per square inch for multiple-cylinder engines having very small fly-wheels and few auxiliaries, and the maximum figure as 3·5 lb. per square inch for single-cylinder engines with heavy fly-wheels and oil and water-circulating pumps, &c. The actual figure adopted must depend in each case upon the general design and type of the engine in question. For convenience of calculation the losses are all given in terms of pounds per square inch of piston area referred to the working stroke only.

The piston friction is the largest item as a rule, and is the most difficult one to deal with. It is assumed that this is proportional to the total pressure on the piston from both fluid and inertia forces combined. It has already been shown that the fluid and inertia forces are not cumulative, but that the total pressures on the piston vary in normal engines from 60 per cent to 80 per cent of the sum of the fluid and inertia pressures. In order to simplify calculation it is assumed that the inertia and fluid pressures are cumulative, and the error is partially corrected by taking a lower figure for the coefficient of friction than actually appears to be the case. Such tests as have been made seem to indicate that the proportion lost in piston friction varies from 3 per cent in gas or petrol engines to $4\frac{1}{2}$ per cent in oil-engines, of the total pressure on the piston referred to the power stroke only. In order to compensate for the error introduced by adding the fluid and inertia pressures together, the proportion of piston friction is taken as 2 per cent and 3 per cent respectively; this presupposes that the total pressure on the piston represents 66·6 per cent of the sum of the inertia and fluid pressures, which is a fair average figure. The formula also presupposes that the ratio of the connecting-rod length to the crank throw is in the neighbourhood of 5 : 1.

The only accurate information usually available is the brake horse-power, and it is only possible to calculate the mean positive fluid pressure from this. The mean pressure of compression usually varies from 20 to 40 lb. per square inch in an explosion engine and from 50 to 60 lb. per square inch in a Diesel engine. This must be doubled in order to arrive at the total fluid pressures. Since this is only a comparatively small variable, and has very little

influence on the total figure, it is convenient to regard the loss due to fluid pressure during the compression stroke as a constant and add it to the constant friction of the piston rings. The value of the constant for both fluid pressure and piston rings is dependent upon the size of the piston and the number and spring tension of the rings. For ordinary gas-engines, using town gas and free from tar or dirt, the constant friction may be taken as equal to 1 lb. per square inch. For small petrol engines it may be taken as 1·5 lb. per square inch. For Diesel engines using a very high compression, 2 lb. per square inch, and for semi-Diesel engines using heavy residual oil or vaporizing oil-engines there is reason to believe that it may be as high as 3 lb. per square inch.

The formula for ascertaining the piston friction in terms of pounds per square inch referred to the power stroke then becomes

$$F = \text{constant} + \frac{\left\{\begin{array}{c}\text{(mean positive fluid} \\ \text{pressure)}\end{array}\right\} + \left\{\begin{array}{c}\text{(mean inertia pressure re-} \\ \text{ferred to power stroke)}\end{array}\right\} \times 2}{100}.$$

The constant must be varied at discretion according to the class of engine, as must also the coefficient of piston friction, which apparently varies between 2 and 3 per cent. depending upon the area of bearing surface and the nature of the fuel. The coefficient of friction of course depends very largely upon the temperature of the cylinder walls, but it may safely be assumed that all tests are carried out at a temperature at which the piston friction is reduced to near its minimum, for the effect which jacket temperature has upon the mechanical efficiency is thoroughly recognized.

In order to calculate the mechanical efficiency of an engine by this means the following information is necessary:

1. The general design of the engine (in order to be able to estimate the value of the constants).
2. The brake horse-power.
3. The bore and stroke.
4. The revolutions per minute.
5. The weight of the reciprocating parts.
6. The effective port area of the valves.

It must, of course, be understood that the above formula refers only to four-cycle single acting engines using trunk pistons, but these form by far the majority of the many types of internal-combustion engine. When an external cross-head is employed there is good reason for believing that the piston friction is reduced to

approximately one-half in spite of the extra weight of reciprocating parts involved. In the case of tandem double-acting engines using separate cross-heads, very high mechanical efficiencies are sometimes obtained if the pencil indicators employed can be relied upon. In such engines a high mechanical efficiency might reasonably be expected, for the ratio of positive fluid pressures to inertia pressures is very high, and the use of external cross-heads will go far to reduce piston friction. The formula is not applicable to two-cycle engines, because in this case the fluid losses are generally much greater and vary widely in different types, and also the influence of the inertia forces is considerably less. Further, in two-cycle engines there is generally a separate scavenge pump whose mechanical efficiency would also have to be ascertained.

The author wishes it to be clearly understood that the above formula is offered rather as a suggestion. Its only justification is that, within the limitations laid down, it agrees fairly consistently with the few accurate results that have been obtained in engineering laboratories. It is very much to be hoped that in the near future those experimenters who have well-equipped laboratories at their command will turn their attention to the investigation of the factors controlling the mechanical efficiency, which influence the performance of an engine to an extent fully as great as the thermodynamic side, upon which much research has already been carried out.

CHAPTER X

VOLUMETRIC EFFICIENCY—COMPRESSION

Volumetric Efficiency.—In any type of internal-combustion engine volumetric efficiency is a consideration of the first importance, for both the commercial and also the thermal efficiency are dependent upon it. For equal temperatures and piston speeds, the mean pressure, and therefore the power of an engine, is dependent almost solely upon the volumetric efficiency, and, since the commercial value of any engine is largely based upon the power it will develop, it follows that, from the commercial point of view, volumetric efficiency is the first consideration. Again, from the point of view of thermal efficiency it is obvious that for a given mean pressure the higher the volumetric efficiency the lower the maximum temperature that can be employed, and therefore the more efficient the engine. In the ordinary accepted type of internal-combustion engine the volumetric efficiency is dependent primarily upon the size of the valves; but it is also dependent to some extent upon the shape of the valve passages, the position of the valves in the cylinder, and the design of the pipe-work. In deciding upon the size of the valves it is necessary to effect a compromise between two conflicting factors, namely, low velocity and turbulence. In order to provide the necessary turbulence, which is essential for rapid and complete combustion, a moderately high velocity of the incoming gases is necessary. On the other hand, the higher the velocity the greater the fluid friction loss and the lower the volumetric efficiency. The actual velocity required to produce the necessary turbulence has yet to be decided, but there seems to be little doubt that 130 ft. per second is ample in any engine in which the combustion chamber is reasonably compact and free from very shallow pockets. In engines in which the valves open directly into the cylinder head a velocity of 100 ft. per second appears to be fully sufficient. With a well-designed valve gear there is very little loss of volumetric efficiency at velocities below 130 ft. per second, but at higher velocities the efficiency begins to fall away fairly rapidly.

The curve illustrated in fig. 28 shows the mean volumetric efficiency of a considerable number of engines of different types. The volumetric efficiency is reduced to terms of standard temperature and pressure, and is therefore considerably lower than that given by direct measurement of the air at ordinary atmospheric temperatures. With an inlet velocity of 100 ft. per second the volumetric efficiency is 74 per cent, and does not vary more than 2 per cent in half a dozen different tests by different experimenters on widely different types of engines. At 200 ft. per second the mean figure for the

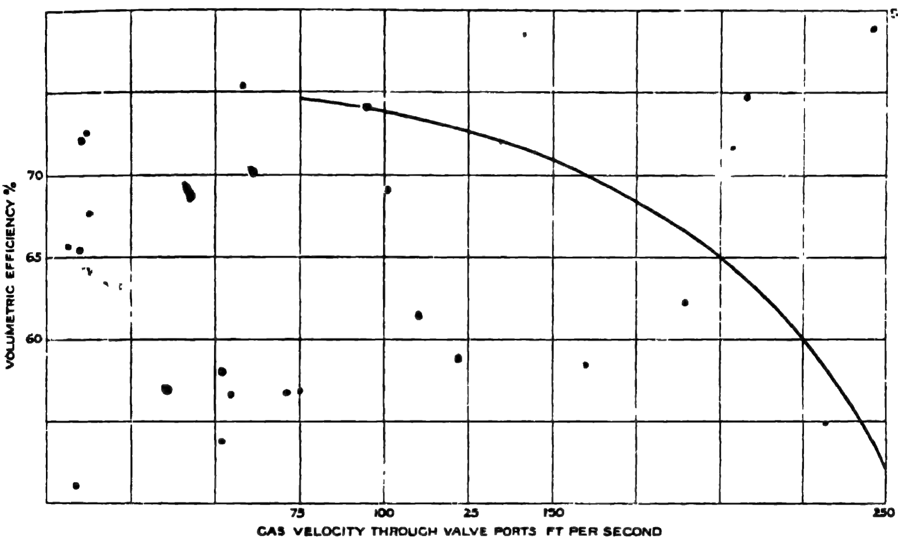


Fig. 28.—Volumetric Efficiency in Relation to Gas Velocity through Valve Ports. Mean of a number of tests on different engines with normal valve settings and poppet valves.

volumetric efficiency is 64 per cent, but this varied from 61 per cent to 67 per cent in six different engines, the variations being due, no doubt, to the different design and position of the valves and passages and the consequent value of the nozzle coefficient. At 250 ft. per second the volumetric efficiency varied from 42 per cent in an engine with shallow side pockets and badly-designed valves and passages to 64 per cent in the case of an engine with valves in the head and no pockets.

Influence of Compression Ratio.—The compression ratio employed has an important bearing on the suction temperature, but very little upon the volumetric efficiency. The lower the compression ratio the lower is the thermal efficiency and the higher the temperature of the residual gases, but, since they are expanded down to atmospheric pressure before meeting the incoming charge,

the difference in temperature between them is small and may be neglected except where a high degree of accuracy is required. Generally speaking, the temperature of the exhaust gases may be taken as approximately 1100° F. under all except abnormal conditions of operations, in which circumstance the actual temperature should be taken.

If the pressure of the residual exhaust gases at the beginning of the suction stroke is only atmospheric then the compression ratio has no influence on the volumetric efficiency, for although, with a low compression, the quantity retained will be greater, yet it must be remembered that when mixed with the incoming charge the residual gases will be cooled, and by contracting will make room for a larger charge, with the net result that the volumetric efficiency is unaffected by the compression ratio so long as the pressure of the gases is not above the atmospheric pressure. In practice, however, it is generally necessary to close the exhaust valve so early that it is almost, if not quite, shut by the end of the exhaust stroke, and the pressure of the gases is generally slightly above atmospheric pressure; under these conditions the compression ratio does have a rather important bearing upon the volumetric efficiency—the higher the compression the higher the volumetric efficiency.

It can be shown that if in any four-cycle internal-combustion engine the pressure within the cylinder at the commencement and end of the suction stroke is exactly atmospheric, then, owing to admixture with the residual exhaust gases, the volumetric efficiency when referred to standard temperature and pressure will not exceed 82 per cent, and that this figure can only be raised either by scavenging, by supercharging, or by cooling the incoming charge to a temperature very considerably below that of the atmosphere. In the case of, say, a petrol engine having a compression ratio of 5 : 1 and a relative efficiency of 67·5 per cent of the air standard, or 32 per cent, the highest mean pressure obtainable with a mixture strength of 100 B.T.U.s per cubic foot, which is the greatest strength consistent with complete combustion, is

$$\frac{100 \times 778}{144} \times \frac{82}{100} \times \frac{32}{100} \text{ lb. per square inch}$$
$$= 142 \text{ lb. per square inch.}$$

This is the absolute limit of pressure obtainable in this engine, assuming that the cylinder is completely emptied of exhaust gases

down to atmospheric pressure at the beginning, and completely filled with fresh charge up to the full atmospheric pressure at the end of the suction stroke. It also assumes the highest practicable compression ratio for a petrol engine, and a very high figure for the relative efficiency when using so strong a mixture. No account is here taken of the lower suction temperature in a petrol engine, due to the latent heat of evaporation of the liquid fuel.

The curves A, B, and C, fig. 29, give the highest mean pressures obtained in the case of A and B, a petrol engine with a com-

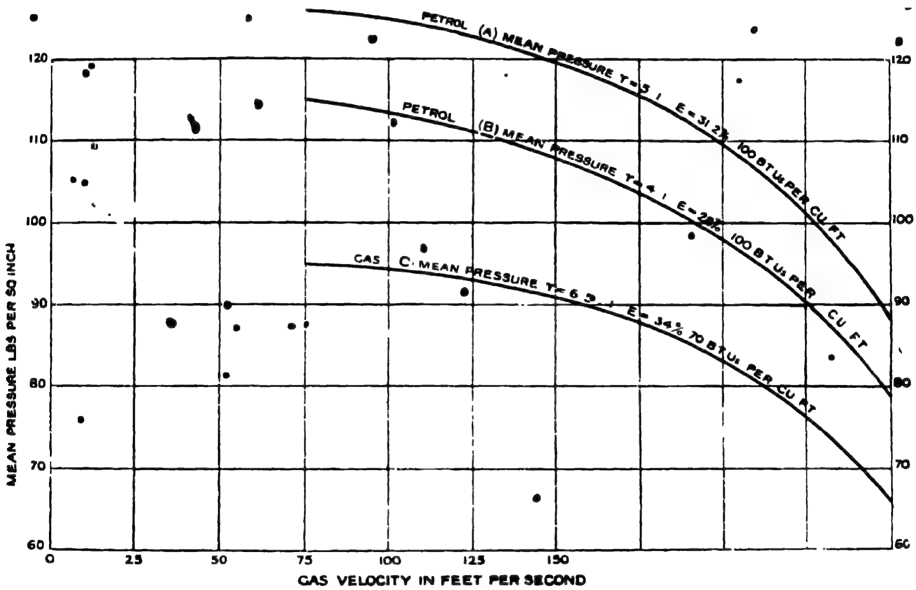


Fig. 29

pression ratio of (A) 5 : 1, and (B) 4 : 1, with gas velocities ranging from 100 to 250 ft. per second through the inlet valves, and with a mixture strength of 100 B.T.U.s per cubic foot as calculated from the mean volumetric efficiency curve (fig. 28): C refers to a gas-engine with a compression ratio of 6·5 : 1, and a mixture strength of 70 B.T.U.s per cubic foot. In both cases it is assumed that the valve timing is normal; that is to say, that the inlet valve does not open until the exhaust valve is closed, so that there is no scavenging, and that the inlet piping is short and of large diameter.

Scavenging.—This can be accomplished by taking advantage of the inertia of the flow of the gases in the exhaust pipe to withdraw some of the residual exhaust gases from the combustion space,

and replace them with either pure air or combustible mixture. To effect this withdrawal it is necessary so to time the opening of the inlet and exhaust valves that both are open together for a short period at the end of the exhaust stroke. In this manner a considerable improvement in the volumetric efficiency can be effected, but it can only be secured when the wave periodicity in the exhaust pipe is in synchronism. If out of phase there is a serious risk of exhaust gases being driven back into the inlet port and then readmitted to the cylinder, so that the proportion of exhaust products is increased instead of being reduced, and the volumetric efficiency diminished in consequence. In engines of the constant-speed type it is possible so to proportion the exhaust pipe that a certain amount of scavenging does take place, but in variable-speed engines it is extremely difficult to do this. When combustible mixture is admitted through the inlet valve this method of scavenging will result in the loss of a certain proportion of unburnt mixture through the exhaust, and is only admissible in cases when it is desired to obtain the maximum possible power without consideration of economy. In some engines, such, for example, as the Premier Gas-engine, positive scavenging by means of an air-pump is employed, with the result that the volumetric efficiency is substantially improved, and mean pressures as high as 110 lb. per square inch can be economically employed even when working with producer gas.

Supercharging.—This can be effected, but to a very limited extent only, by taking advantage of the inertia of the gases in the induction pipe to raise the pressure within the cylinder at the end of the suction stroke to a value above atmospheric. It can also, of course, be accomplished mechanically by means of an air-pump.

Exhaust Back Pressure. So far, the effect of restricting the free flow of the exhaust gases has not been considered, and, except in the case of very badly designed exhaust silencers, it need not be taken into consideration, because, as a general rule, owing to the exceedingly high velocity of the exhaust gases when first released, their inertia is so great as to overcome the resistance of any reasonably well designed silencer without throwing any serious back pressure upon the piston, or appreciably influencing the volumetric efficiency.

Examination of light-spring indicator diagrams generally reveals the fact that the exhaust pressure falls to almost atmospheric pressure at about the middle of the exhaust stroke, but rises to

about 2 lb. per square inch above atmospheric at the end of the stroke owing to the early closing of the exhaust valve, except in such cases when the valves are set for scavenging. It is only the pressure at the end of the exhaust stroke which has any influence upon the volumetric efficiency, and this is nearly independent of the average pressure during the stroke. Exhaust back pressure may considerably increase the negative work on the piston, but it will have very little influence on the volumetric efficiency, unless, as is too often the case, the exhausts from multi-cylindered engines are so connected that one cylinder exhausts into another just at the beginning of its suction stroke; when this occurs it is obvious that the clearance space receives a charge of highly-heated gases immediately before the commencement of the suction stroke, and this naturally has the worst possible influence upon the volumetric efficiency. The question of pipework generally, and its effect upon charging and exhausting, is dealt with at greater length in another chapter.

Influence of Compression on Thermal Efficiency.—Althoug theoretically, the efficiency of an internal-combustion engine is dependent upon the compression ratio, in practice this statement requires considerable revision, for although the theoretical efficiency increases with increase of compression, the mechanical efficiency decreases, so that a point is reached at which any further increase in the compression pressure is counteracted by increased mechanical losses, and the net gain is *nil*.

In fig. 30, curve A represents the air-standard efficiency for compression ratios, varying from 2 : 1 up to 14 : 1, and curve B 71 per cent of the air standard, and is the actual indicated efficiency which might be expected from first-class engines running on the constant-volume or explosion cycle—71 per cent of the air standard being the highest proportion realized in first-class engines working with a comparatively weak mixture, with a mean effective pressure of about 80 lb. per square inch on town gas, and corresponding pressures on other fuels. It will be noticed that with compression ratios above 7 : 1 the increase in the indicated thermal efficiency is comparatively small.

It has already been shown that the mechanical efficiency of any ordinary single-acting trunk-piston engine is dependent, other things being equal, upon the compression ratio, since it is this which determines the maximum pressures and therefore the weight of the working parts. The curve C, fig. 30, gives the highest mechanical

efficiency generally obtained with various compression pressures in single-acting four-cycle engines of present-day design. The piston speed in each case being taken as 750 ft. per minute, curve D gives the actual or brake thermal efficiency obtainable. It will be noticed that curve D reaches a maximum of 36 per cent when the compression ratio is 11 : 1, and declines slightly when the compression ratio is still further increased. In practice the explosion or constant-volume cycle cannot be employed when the compression ratio

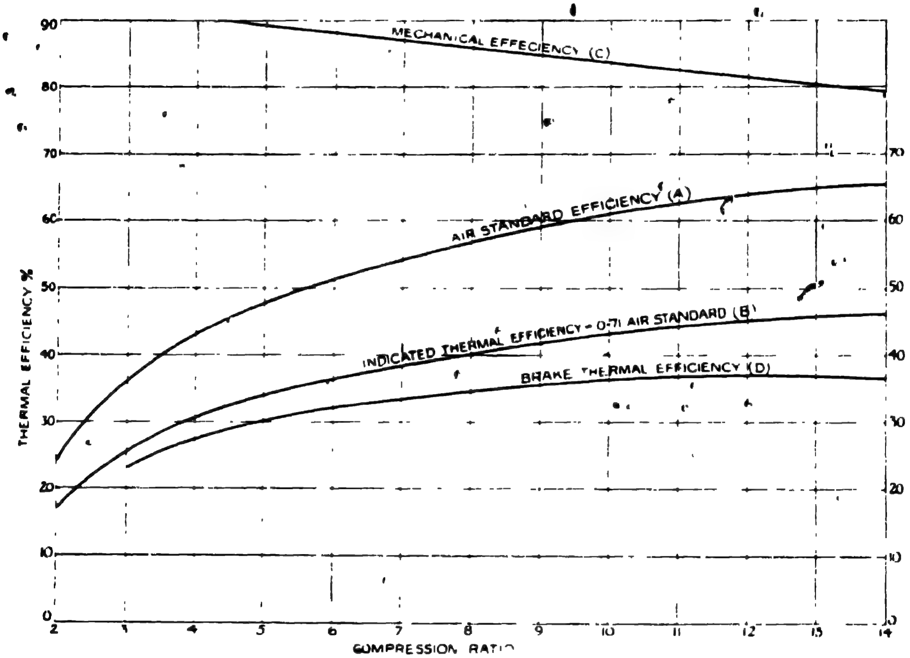


Fig. 30.—Curve showing Influence of Compression Ratio upon Efficiency

exceeds about 7·5 : 1, because the temperature of compression becomes so high as to cause premature ignition; hence at the higher compressions a different and less efficient heat cycle must be employed, and that is why engines of the semi-Diesel type having a compression ratio of about 10 : 1 do not show brake thermal efficiencies as high as 36 per cent. The curves of mechanical and brake thermal efficiencies apply only to single-acting trunk-piston four-cycle engines. When double-acting cylinders are employed, or when an external cross-head is used, the mechanical efficiency is considerably higher, and the conclusions with regard to the maximum compression ratio require modification. The curve D is interesting, in that it shows how comparatively little advantage

is gained in practice by the use of very high compression, and that there is no gain whatever in using a compression ratio higher than about 11·5 : 1, corresponding to a compression pressure of about 360 lb. per square inch, unless some means be found for increasing the mean pressure to a figure considerably in excess of that at present obtained.

Pre-ignition.—In the case of explosion engines the degree of

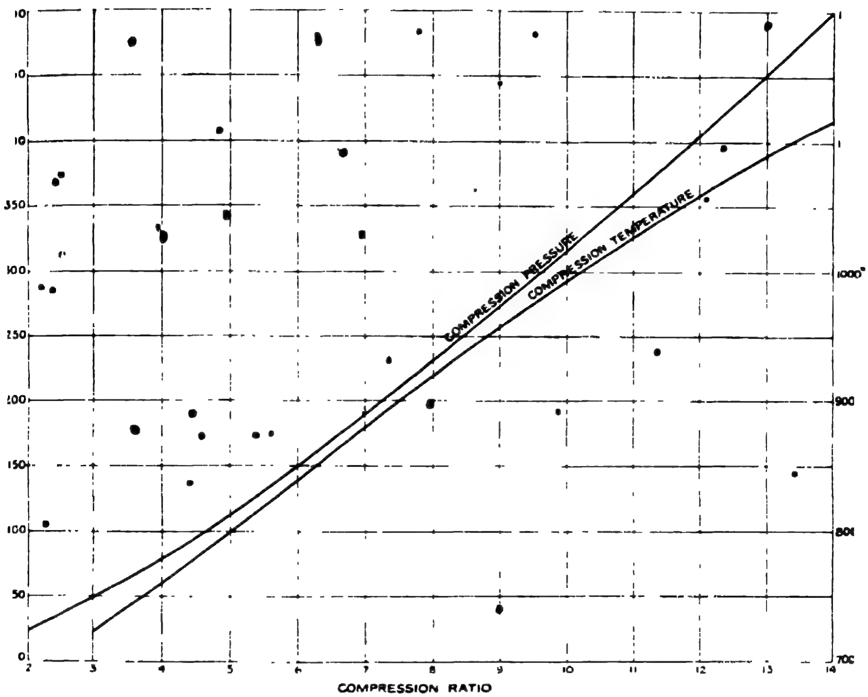


Fig. 31. —Temperature and Pressure of Compression. $(P \times v)^{1.35} = C$. Initial temperature varying according to compression ratio

compression is limited by the ignition point of the fuel, for it is obvious that to guard against the danger of pre-ignition the maximum temperature of compression must be well below this point. Professor Hopkinson has found that the ignition point of town gas is about 1350° F. when compressed with air to a pressure of 150 lb. per square inch, and is about 1550° F. at atmospheric pressure; that is to say, he found that pre-ignition occurred just before the end of the compression stroke if any point within the cylinder exceeded 1350° F., and that the charge ignited as it entered when the temperature rose to 1550° F.

The curves, fig. 31, give the pressure and temperature of the

gases at the end of the compression stroke for different compression ratios. These curves are calculated on the assumption that the value of γ is 1·35, that the temperature of the gases at the commencement of the compression stroke is proportional to the quantity of exhaust gases retained in the clearance space, and ranges from 300° F. with a compression ratio of 4 : 1 down to 160° F. with a ratio of 14 : 1, and that the pressure at the commencement of the compression stroke is exactly atmospheric.

Engines of the explosion or constant-volume type use the highest compression which is compatible with freedom from pre-ignition, and this in practice generally ranges from 3 to 7·5 : 1, depending upon the fuel. With a 3 : 1 compression ratio, as frequently used in paraffin engines, the maximum temperature of compression is 720° F., while with a compression ratio of 7·5 : 1 the temperature is 900° F. Both these temperatures are considerably below the normal ignition point of the fuel, but it is necessary to provide a very wide margin, because it is not the mean temperature of the whole bulk of the mixture that has to be considered, but the highest local temperature at any one point. This may be raised to a very much higher figure than the mean by the presence of such uncooled parts as the exhaust valves and igniter points, and also by particles of carbon which are deposited on the walls of the combustion space and the piston head, and which, being poor conductors of heat, are liable to reach a very high temperature. There is, moreover, another cause of premature ignition which is not generally recognized, but which is none the less very prevalent, namely, that due to particles of lubricating oil coming in contact with hot surfaces, such as the head of the exhaust valve or the centre of the piston; under these conditions the oil will be decomposed or "cracked", throwing down a deposit of carbon and liberating free hydrogen which has a very low ignition point. Professor Hopkinson has found that ordinary gas-engine oil will crack and cause pre-ignition if dripped on the head of the exhaust valve when the temperature of the surface exceeds about 800° F. In order to guard against this source of pre-ignition, which is more prevalent in horizontal than in vertical engines, the exhaust valve should be placed at the lowest point in the cylinder, so that it will act as a drain and prevent the accumulation of lubricant; this arrangement not only minimizes the risk due to pre-ignition from cracking of the oil, but it also, of course, reduces the carbon deposit on the cylinder walls and piston head, most of

which is due to decomposition of the lubricating oil rather than of the fuel.

Explosion engines using paraffin, and also those using coke-oven gas, which has a high proportion of hydrogen, are handicapped by the low ignition point of the fuel. Paraffin engines are still further handicapped by the necessity for pre-heating the combustible mixture in order to vaporize it; for this reason such engines can, as a rule, only use compression ratios of about 3 to 3·7 : 1, and since the pre-heating of the fuel also results in a considerable reduction in the volumetric efficiency, it follows that the mean pressure and therefore the mechanical efficiency are reduced. As a result of this the brake thermal efficiency of such engines is limited to about 21 per cent. In practice the efficiency is generally lower, owing to defective vaporization and combustion. In order to make possible the use of a higher compression ratio, and so improve the efficiency of paraffin engines, various devices are resorted to with the object of preventing pre-ignition, either by reducing the compression temperature or preventing admixture between the component parts of the combustible mixture until near the end of the compression stroke.

In order to reduce the compression temperature, it is a very common practice to admit a small quantity of water in the form of a finely-divided spray into the cylinder during the suction stroke. This water is then evaporated during the compression stroke, and takes up a portion of the heat of compression, thus enabling a higher compression ratio to be employed without increasing the compression temperature. By this means the efficiency can be very substantially improved, and a compression ratio as high as 4, or in some cases 4·5 : 1, can be employed, which would allow of a brake thermal efficiency of about 24 per cent if the fuel were completely vaporized.

In experimental engines compression ratios as high as 8 : 1 have been employed with very favourable results, but since such engines depend entirely on the water injection for their immunity from pre-ignition under all loads, and are liable to pre-ignitions of an extremely violent and dangerous nature if the water supply is only momentarily interrupted, they can hardly be regarded as commercial successes. The effect of the water, however, is slightly to reduce the efficiency, because of the greater specific heat of the steam, but this is more than compensated for by the gain due to the higher compression ratio. There is a mistaken idea prevalent that the admission of water in itself raises the efficiency of an engine; such an idea is entirely erroneous, the only advantage of using

water is to make possible the use of a higher compression. If an engine is designed with a sufficiently low compression to run without water injection, then the sole effect of adding water will be to lower the efficiency. Although the employment of water injection permits of a higher compression ratio and a higher brake thermal efficiency, its use is not altogether desirable, for, in the first place, the quantity of water has to be carefully regulated according to the load; if too little is admitted pre-ignition may occur, and if too much it will not be completely evaporated, will play havoc with the lubrication of the piston, and cause corrosion, owing to its combination with the sulphur which is always present to a greater or lesser degree in oil or gaseous fuels. Again, the water itself must be soft and pure or there will be an excessive deposit of lime in the cylinder.

Another method enabling higher compression ratios to be employed is to admit the mixture in two sections, so to speak, one section consisting of air so rich in vapour as not to be inflammable and the other of pure air. The two sections are admitted (1) into a bulb separated from the remainder of the combustion space by a restricted neck, (2) into the main body of the cylinder. During the compression stroke the air in the main body of the cylinder is driven into the bulb until, at the end of the stroke, almost all the air has entered, and a combustible mixture of the correct proportion is formed. This arrangement permits of only a very slight increase of compression, for the line of demarcation between a combustible and incombustible mixture is not so fine as to make it possible to ensure that the mixture in the bulb shall be not inflammable until the extreme end of the compression stroke. One of the principal advantages of this system is that the greater part of the charge is not pre-heated, and thus the suction temperature is lower.

In coke-oven gas-engines the use of water injection is practically inadmissible on account of the very large percentage of sulphur always present in this gas, and very great care must be taken to ensure that no water shall under any circumstances get into the cylinder, or corrosion will take place very rapidly. With coke-oven gas-engines, however, it is generally possible to use a compression ratio of from 4·5 to 5·5 : 1 without serious risk of pre-ignition, and since this gas is of high heat value and produces a high mean pressure, the mechanical efficiency is high, and a brake thermal efficiency of 29 per cent to 31 per cent is obtainable. In modern coke-oven gas-engines of large size it is now becoming common practice to

admit a certain proportion of cooled inert exhaust gases during or after the suction stroke, in order further to dilute the hydrogen and enable a higher compression ratio to be employed.

In gas-engines using blast furnace and some forms of producer gas a compression ratio as high as 7·5 : 1 can be employed, and still higher ratios would be safe were it not that even in such gases the proportion of hydrogen may rise considerably above the normal. Also such gases are liable to contain a considerable amount of dust, the particles of which may become incandescent and cause a local rise of temperature above the ignition point of the fuel.

In engines of the Diesel or constant-pressure type the heat of compression is relied upon to ignite the oil, and it is important to ensure that such ignition shall take place with certainty, even at starting, when the temperature of the cylinder walls and the suction temperature are at a minimum. A failure to ignite may result in a severe and dangerous pre-ignition, because a considerable proportion of the fuel will be retained in the cylinder during the succeeding compression stroke, and will be vaporized and ignited before the dead centre. It follows, therefore, that in such engines the compression must be carried to such a degree that the ignition temperature of the fuel is exceeded by an ample margin. The ignition temperature of the various fuel oils under compression varies considerably, but in the case of petroleum fuels appears to be in the neighbourhood of 800° F. In most Diesel engines a compression ratio of from 13 to 14:1 is employed, giving a compression temperature of about 1080° F. to 1100° F. under normal running conditions, starting with a suction temperature of 166° F. At starting, however, the suction temperature is only about 50° F., and sometimes even lower, so that the final temperature at the end of compression will be only 825° F., which provides a very small margin. If a lower compression ratio than 13:1 be employed, it is advisable to heat the cylinder jackets before starting in order to ensure a sufficiently high compression temperature. Reference to the curve, fig. 30, will show that a compression ratio of 13:1 is actually too high for the best brake thermal efficiency. With this ratio the weight of the reciprocating parts, which must be strong enough to withstand pre-ignition, will be so great that the mechanical efficiency (exclusive of the blast air-pump) will generally not exceed about 80 per cent, with a piston speed of 750 ft. per minute and a mean pressure of 100 lb. per square inch.

In engines of the so-called semi-Diesel type, in which the fuel

is ignited partly by the heat of compression and partly by heat added from uncooled surfaces, any degree of compression can be employed, for it is clear that the necessary temperature can be produced by supplementary heat added from the uncooled surfaces, such engines can therefore use whatever compression gives the best thermal or commercial efficiency. The commercial efficiency of an engine depends largely upon its size, for the larger the engine the greater is the ratio of fuel cost to first cost; it follows, therefore, that the larger the engine the more important does the thermal efficiency and the less important does the first cost become. The first cost of an engine is governed very largely by the compression ratio, which, other things being equal, decides the weight and therefore the cost of an engine. As a general rule, the highest commercial efficiency of a large engine of say 100 B.H.P. per cylinder is reached when the compression ratio is about 9 to 10 : 1, and of a small engine of say 10 B.H.P. per cylinder when it is about 6 to 7 : 1; and since with the semi-Diesel engine either compression ratio can be employed, it would follow that the most suitable ratio would be anything from 6 to 10 : 1 according to size. There are, however, reasons, which will be dealt with later, why semi-Diesel engines should not have too low a compression; and although compression ratios of even less than 6 : 1 are sometimes employed, their use entails a good deal of trouble, owing to the large quantity of added heat required and the difficulty of regulating it satisfactorily.

Suction Temperature.—Both the volumetric efficiency and the compression temperature are largely dependent upon the suction temperature, that is, the mean temperature of the gases within the cylinder at the end of the suction stroke. It is obvious that if the suction temperature can be reduced, both the volumetric efficiency and the compression ratio can be increased, and very substantial advantages gained thereby. The suction temperature is governed chiefly by the proportion of exhaust products retained in the combustion space, which mix with and heat the incoming charge; and also by the heat given up to the entering gases from the hot walls of the cylinder, piston, &c. Unfortunately the suction temperature is exceedingly difficult to measure, and very few experimenters have any data regarding this point. Professor Hopkinson, in his tests on a Crossley gas-engine with a compression ratio of 6·37 : 1, found that the suction temperature was 220° F. when the temperature of the outside air was 60° F. The amount of heat added during the admission period was therefore sufficient to raise the temperature of

the air and gas through 220° – 60° F., or 160° F. Of this temperature rise he estimated that the hot exhaust gases were responsible for approximately 133° F., and the hot walls of the cylinder piston and valves for 27° F.

Professors Coker and Scoble found that the suction temperature of a small gas-engine with a compression ratio of 4·85 :1 was 257° F.; presumably the heat taken up from the walls would not account for more than about 40° F. in this engine, leaving 217° F. temperature rise due to admixture with the exhaust products.

These two results represent, apparently, all the data available at the present time, and since so much depends upon the suction temperature it is obvious that there is a crying need for further investigation upon this point.

In four-cycle engines, in which the air is drawn directly into the cylinder from the outside atmosphere, it is clear that its temperature cannot easily be reduced below the normal temperature of the atmosphere; but in two-cycle engines, in which the air is delivered to the working cylinder from a pump, which has done work upon it and raised its temperature, a very considerable advantage can be obtained by passing it through an intercooler on its way to the working cylinder.

CHAPTER XI

DETAILS OF DESIGN

The importance of reducing the weight of the reciprocating parts of an internal-combustion engine to the lowest possible limit has already been emphasized, and is fully realized by the designers of high-speed petrol engines, who have exercised extraordinary ingenuity in producing light pistons. It does not, however, appear to be properly understood by the designers of engines of other types.

From the foregoing considerations it is obvious that the present design of piston is susceptible of considerable improvement, both as regards reduction of weight and lubrication. The fact that the piston friction can temporarily be reduced to only about one-half of its normal value, or even less by profuse lubrication, suggests the advisability of using a separate cross-head to take thrust, which could be kept cool and profusely lubricated with thin oil. In large engines this is frequently done, and results in a marked improvement in the mechanical efficiency despite the increased reciprocating weight.

The main objection to the use of a separate cross-head is the increase in the height or length of the engine, and the consequent increase in cost, weight, and, in a horizontal engine, in floor space. It would seem, however, that a sort of combination piston and cross-head could be made without increasing the length of the present type, but so designed that its two functions, (1) as a piston proper, and (2) as a cross-head, could be treated separately.

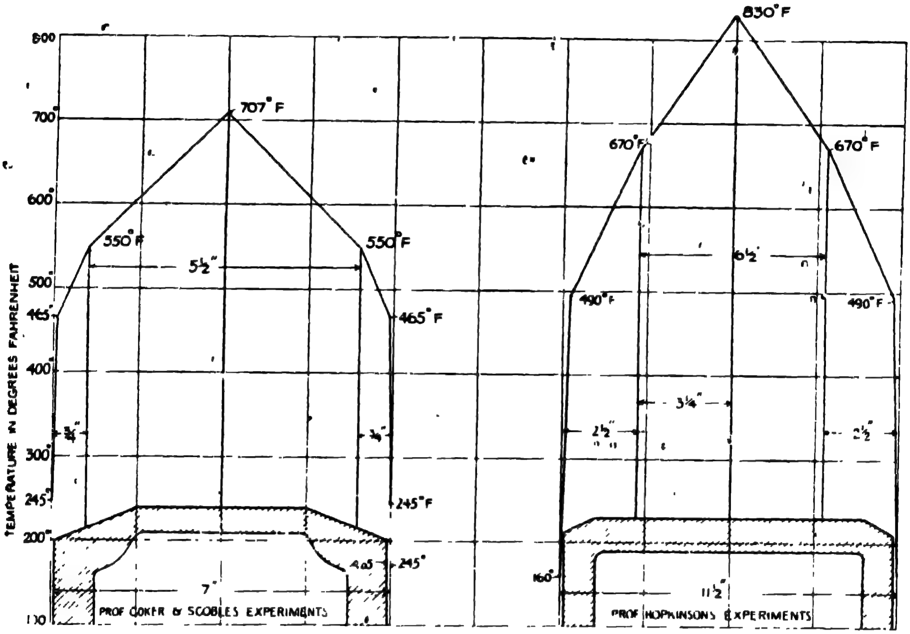
Thermal Considerations of Piston Design.—So far the design of the piston has been considered only from the point of view of mechanical friction, but it is also necessary to consider it from the point of view of heat dissipation. In all small gas and oil engines the pistons are uncooled, for water or oil cooling of the pistons presents serious mechanical difficulties, besides adding greatly to the weight. It is, therefore, desirable to avoid the necessity for this as far as possible. Since the heat from the

centre of a piston of normal design can only be dissipated either by conduction to the cylinder walls or by radiation and convection of the air currents on the inside, it follows that the centre gets very much hotter than the circumference, which is in contact with the cylinder walls. This great difference of temperature, and, in consequence, of expansion between the centre and the circumference, introduces serious stresses, which, in large cast-iron pistons, become so severe that the metal is in some cases strained beyond its elastic limit. Professor Hopkinson has shown that the stresses due to the temperature gradient across the head of a comparatively thin cast-iron piston may, in the case of a piston over 12 in. in diameter, running at maximum load with rich gas, be so great as to cause cracks from unequal expansion alone and without the application of any external pressure. As a matter of fact, uncooled pistons very much larger than 12 in. diameter are in successful use, but the metal requires to be very carefully selected. Such pistons are very thick and heavy, and are not generally used in conjunction with very rich gas.

Besides the stresses set up by the differences of temperature across the head of the piston, there is also the danger that the centre will become so hot as to cause premature ignition, for the temperature of the centre of the piston varies as the square of the diameter if the thickness is constant and directly as the diameter if the thickness is in proportion to the diameter. This trouble is particularly common when the piston becomes encrusted with carbon, which, being a poor conductor of heat, gets considerably hotter than the metal. Numerous experiments have been made by means of thermocouples by Professors Hopkinson, Coker, and others, to ascertain the difference in temperature that exists in different parts of the head of a piston. The results of these experiments are illustrated in fig. 32. It would seem that an engine using a piston rod and crosshead would be in a better position from this point of view, since a considerable proportion of the heat imparted to the centre would be conducted along and dissipated by the piston rod, so that in such engines larger pistons could be used without the necessity for water-cooling. Quite recently certain alloys of aluminium containing about 12 per cent of copper have been suggested for the pistons of aeroplane engines and also for racing motor-car engines. These alloys have been surprisingly successful in experimental engines. Should their success be fully established it is obvious that a very great advance will have been made, for not only is the weight of these alloys very

much less than that of cast iron or steel, but what is even more important, their conductivity is very much greater, so that it seems quite possible that very much larger uncooled pistons could be safely employed, for, even if for the sake of conductivity the thickness of the head were increased by 100 per cent, they would still be much lighter than cast iron.

The piston rings themselves do not call for any particular comment. Although numerous types of rings have been tried with



Temperature of cylinder jacket water, 200° F
,, inner surface of liner, 245 F.
,, piston centre of liner, 707 F.
,, outer edge, 465° F
Difference between outer edge of piston and inner face of liner, 220 F

Temperature of jacket water, 100° F.
,, inner surface of liner, 160° F.
,, piston centre, 830 F.
,, outer edge of piston, 490° F.
Difference of temperature between outer edge of piston and inner face of liner, 330 F.

Fig 32

more or less success, the ordinary split eccentric Ramsbottom type is still the general favourite, and indeed, if carefully made from a suitable grade of cast iron, there is very little fault to be found with it, while its simplicity and low cost of production compared with other and more fancy types are very much in its favour. Such rings should be ground both on the sides as well as on the working face.

There is much difference of opinion both as to the most desirable stiffness of spring tension and as to the width of the ring. With regard to the former, much must necessarily depend upon

the accuracy of workmanship. When the liner is accurately bored or ground, and the piston so designed that there is little distortion due either to expansion or the transmission of pressure through the walls of the piston carrying the rings, there can be little doubt but that very light rings may be used with advantage, for not only does the use of light rings lessen the friction losses, but it reduces also the wear both of the rings themselves and of the cylinder walls. On the other hand, light rings naturally take longer to bed, and will not accommodate themselves so readily or so quickly to lack of roundness in the liner, &c. When the workmanship is of high class, it is found that a spring tension of about 4 to 5 lb. per square inch is sufficient to ensure gas tightness. The spring tension in terms of pounds per square inch on the surface of the ring is, of course, independent of width, and is controlled solely by the radial thickness of the ring. The lower limit of spring tension is reached when the friction against the side of the groove due to the third pressure on the exposed side of the ring exceeds the spring tension, so that the ring becomes locked in its groove, and is unable to expand during the period of high gas pressure. This again depends to a large extent upon the clearance between the lands on the piston and the cylinder walls which controls the area of exposed surface—the smaller the clearance, therefore, the less spring pressure is required. It depends also, to some extent, upon the clearance of the crosshead portion of the piston.

As to the width of the ring, there can be little doubt but that the narrower it is the better, on the score both of friction and wear in the ring-grooves; for in regard to wear, the area of the surface in the ring-grooves is governed solely by the radial thickness, while both the inertia and the friction drag of the rings is directly proportional to the width. It follows, therefore, that the loadings on the sides of the groove and therefore the wear will increase directly in proportion to the width of the ring. Wear in the ring grooves is perhaps one of the most troublesome features to contend with, for once a ring becomes slack in its groove its reciprocating motion therein converts it into a very efficient pump for pumping oil up into the combustion chamber.

In practice the minimum width of a piston ring is determined largely by manufacturing considerations and its fragility in handling, but, generally speaking, a ring should be as narrow as is consistent with ease of handling and of manufacture.

From the foregoing considerations it would appear that the

main features to be aimed at in the design of a piston for an internal-combustion engine are:—

1. It should be as light as possible.

2. That portion of the piston which is required to perform the functions of a cross-head should be designed as such, and due attention paid to the selection of suitable material, lubrication, and adjustment for wear.

3. That portion of the piston which is required to function as

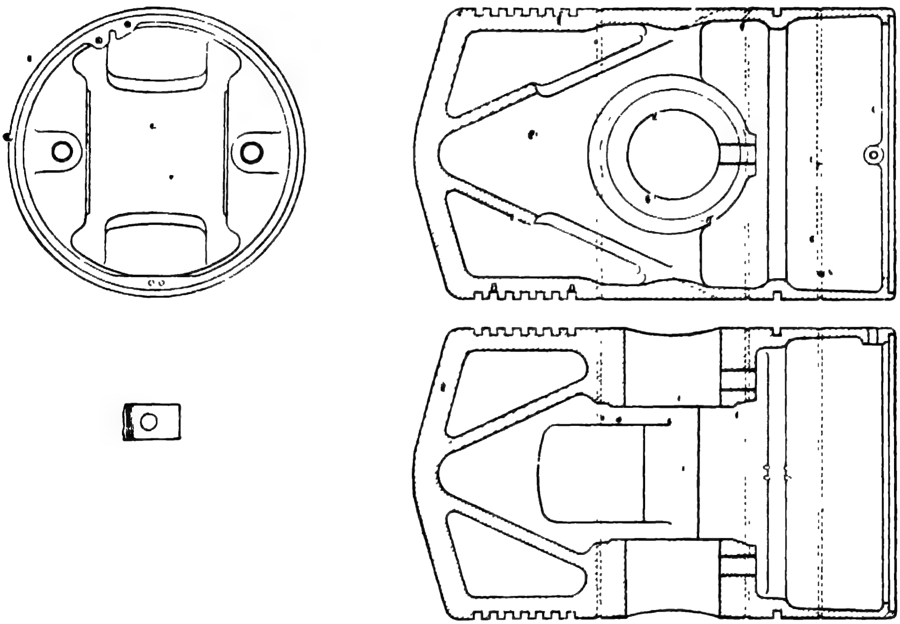


Fig. 33.—Typical Gas-engine Piston (Ruston Proctor)

a piston pure and simple should be designed so as to dissipate the heat in the most efficient manner; it should combine the necessary strength with the minimum weight, and should be made of a material sufficiently elastic to withstand wide variations of temperature without serious loss of strength or the setting up of dangerous internal stresses.

4. Provision should be made for the free escape of any gases that may pass the piston rings, so that they shall not interfere with the lubrication of the cross-head portion of the piston.

5. Provision should be made for the free ventilation of the piston proper.

6. The piston proper should not receive any of the thrust due to the angularity of the connecting-rod.

7. For the sake of reducing the weight, the pressure upon the piston should be transmitted as directly as possible to the connecting-rod.

Figs. 33, 34, 35, and 36 are shown to the same scale—33 represents a typical gas-engine piston, 34 a typical petrol-engine cast-iron piston, 35 a special steel petrol-engine piston as used for racing motor-car and aero engines, and 36 a gas-engine piston of a design proposed by the author to fulfil the requirements set forth above.

Considering first the two pistons shown in figs. 33 and 34, the conditions under which they work are very similar. The petrol-engine piston is subjected to much higher temperatures, but is con-

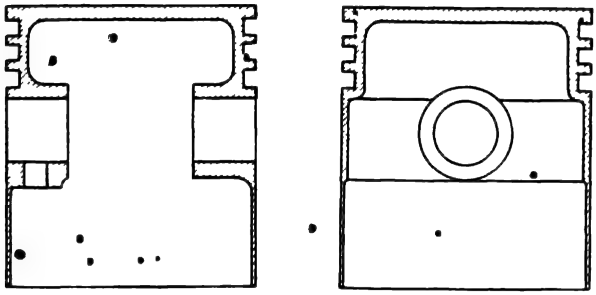


Fig. 34

siderably the smaller of the two, though the actual power output per piston is but little different in the two cases. A glance at these two designs will show that the designer of the petrol engine has realized the importance of reducing the weight of the piston, and also that he has confidence in the ability of the foundry at his disposal. Both pistons are of the same type, the principal difference being in the disposition of the metal.

The piston shown in fig. 35 is machined from a solid steel billet, and its extreme costliness is an indication of the importance which its designer attaches to light weight. In this design an attempt has been made to convey the pressure direct from the centre of the piston head to the connecting-rod, and at the same time to dissipate some of the heat from the centre, by the provision of a central pin connecting the gudgeon pin with the piston head. By doing this the designer is enabled to reduce the thickness both of the head and walls to a considerable extent, and so to produce an exceedingly light piston and gudgeon pin. The design is most effective, and its use enables the particular engine to which it is fitted to run at an extraordinarily high speed without undue vibration or friction loss. This design is

ingenious, too, from a manufacturing point of view, for it is obvious that, if the hole for the gudgeon pin is drilled first, and the hollow part of the piston machined out afterwards, leaving the central pin, then the end of this pin can be relied upon to bear truly upon the gudgeon pin without any intricate fitting. The objection to this

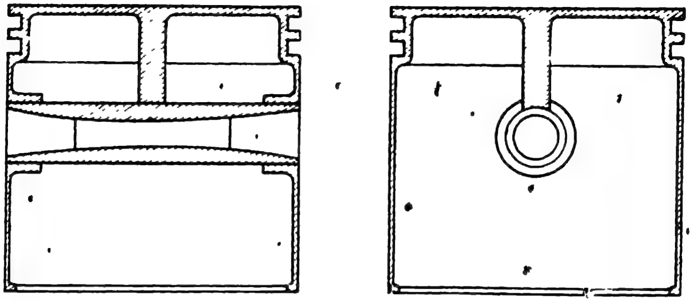


Fig. 35

design is that the cross-head portion of the piston is of steel, an unsuitable material from the point of view of lubrication and wear; nevertheless it is an exceedingly clever piece of design.

The design shown in fig. 36 is intended by the author to fulfil the various conditions enumerated above. It is not suggested that it would be suitable for small engines. It will be seen that

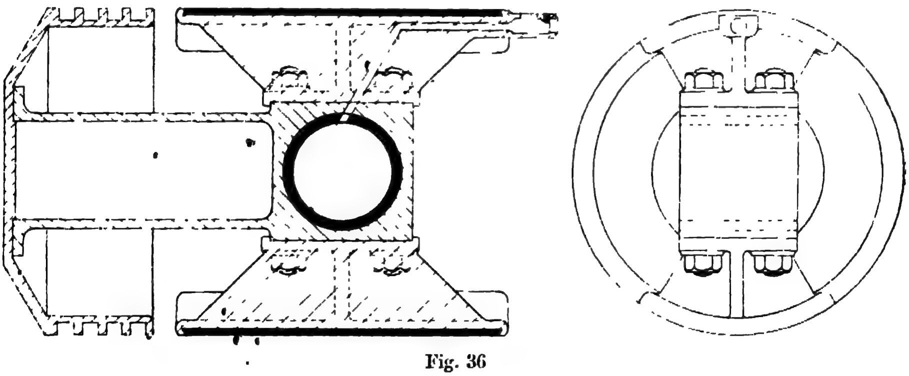


Fig. 36

the design is a composite one, embodying a piston proper, and a cross-head connected by a short hollow piston rod. The piston proper is of forged steel or possibly of aluminium alloy. It is machined considerably smaller than the cylinder, and is not intended to bear upon it at all. Since the pressure is transmitted directly from the head of the piston to the connecting-rod along the large-diameter hollow piston rod, it follows that the head can be kept

very thin and light. The heat is dissipated partly through the piston rings, direct to the cylinder walls, and partly along the hollow piston rod, which latter may be provided with a few small radiating flanges to assist the cooling. The cross-head portion of the piston is designed upon the orthodox lines of an ordinary steam-engine cross-head, with adjustable slippers. The slippers, which are made as light as possible, may be faced with anti-friction metal.

Such a piston could probably be made to weigh less than half as much as an ordinary cast-iron piston. Its cost, of course, is considerable, but when it is considered that its use should enable an engine to run at about 40-per-cent higher speed than is possible with an ordinary type cast-iron piston, with the same or less friction loss, the extra cost may be fully justified.

The importance of reducing the weight of the piston from the point of view of balance should not be forgotten, but as this side of the question is dealt with under the head of Balancing, it is unnecessary to reiterate it here.

CHAPTER XII

VALVES AND COMBUSTION HEADS

In the arrangement of the combustion head and valves the designer has to compromise between a considerable number of conflicting requirements, and before deciding upon any final arrangement, he must take into account all the various purposes for which his particular engine is likely to be used, and make up his mind whether the chief aim is thermal efficiency, volumetric efficiency, mechanical strength, good governing over a wide range of load and speed, or low cost of production. His choice of arrangement will also depend upon the size of the engine, and whether it is intended to manufacture it in single-cylinder units, multiple-cylinder units, or both. Before discussing the various arrangements now in use it will be well first to tabulate the conditions which the ideal combustion head and valves should fulfil from the various points of view.

1. **Thermal Efficiency.**—The combustion head and valve disposition should be such as to present the smallest possible surface to the gases, in other words should be hemispherical or nearly so.

2. **Volumetric Efficiency.**—The valves should be of such a size that the mean gas velocity through them shall be between the limits 100 to 130 feet per second, and the area of the passages leading to them should be greater than that of the valve ports. The gas passages should be free from any sharp angles or sudden changes of area, and the disposition of the valves should be such that the pipework leading to and from the cylinder will satisfy the same conditions. The valves should be arranged in the combustion head in such a way that the free flow of the gas is not restricted by the too close proximity of the walls, in other words they should open direct into the cylinder and not into a restricted pocket or chamber.

3. **Mechanical Strength.**—The combustion head must be designed with due regard both to strength and the rapid dissipation of heat, and the disposition of the valves must not be such as either to seriously weaken the head or obstruct the free flow of the circulat-

ing water. The thickness of metal throughout the combustion head should be as nearly uniform as possible, both to ensure a sound casting and to prevent wide differences in temperature with the consequent serious danger of cracking. It is, moreover, desirable that the valves should be so arranged that, in the event of the breakage of a valve stem or cotter, they cannot fall into the cylinder.

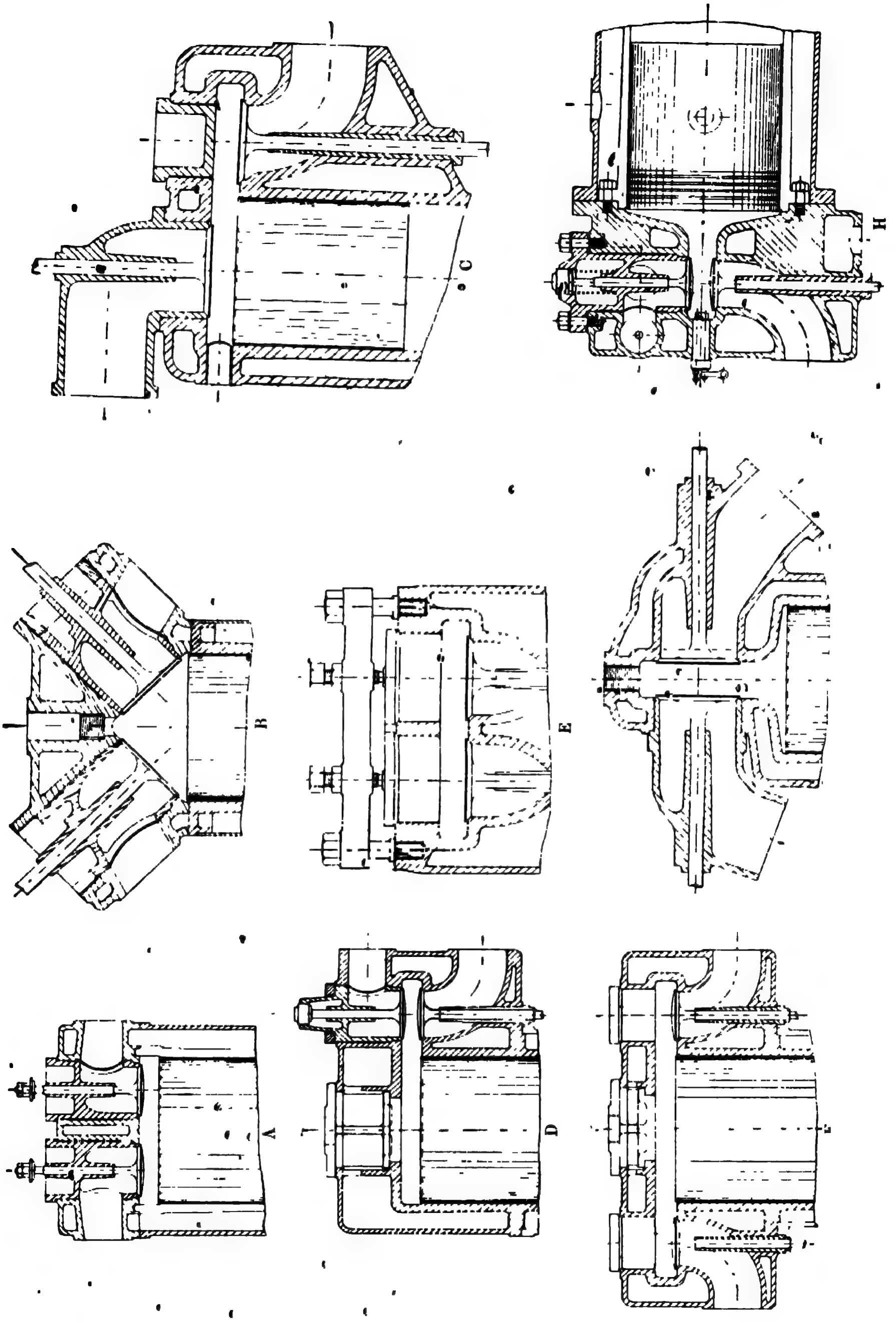
4. **Good Governing.**—It must be remembered that in a four-cycle engine the products of combustion are not entirely expelled, and that the clearance space is left full of exhaust gases at, or slightly above, atmospheric pressure.

In cases where the method of governing by throttling the mixture is employed, and this is almost the only practical method for gas-engines which are required to deal with very widely varying loads, the proportion of fresh charge drawn in may be so small, compared with the residual exhaust gases, that, if it mixes thoroughly with them, it becomes so diluted as to be non-inflammable. To obviate this a certain degree of stratification is essential, and it is desirable that the inlet valve and the igniter be separated to some extent from the main body of the cylinder, in order to ensure that there shall always be a small proportion of fresh gas in close proximity to the latter, otherwise the engine will not fire regularly on light loads. This condition applies only to explosion engines. It is directly opposed to the conditions governing thermal and volumetric efficiency.

5. **Cost of Production.**—The combustion head should be designed in such a way as to keep down the cost of manufacture, and the valves should be arranged so that they can be operated in the simplest possible manner. The disposition of the valves will depend upon whether the cylinders are to be horizontal or vertical, the number of cylinders, and, to some extent, upon their size. Although horizontal valves may be used with satisfaction in small engines, their employment in large engines, when their weight becomes appreciable, is said to be undesirable.

A number of different valve arrangements favoured by various designers are illustrated in fig. 37, and it is interesting to examine these and see how far they conform to the conditions set down above.

The arrangement illustrated at A is a very popular one for vertical engines, especially Diesel engines. It is also sometimes employed for stationary gas-engines, and occasionally for motor-car work. It fulfils the first and second conditions fairly well, but, where very high speeds are required, it has the disad-



vantage, that it is not possible to fit valves large enough for the best volumetric efficiency, unless the top of the cylinder be enlarged, which is detrimental because it increases the area of surface exposed

to combustion. From the mechanical point of view it is a bad arrangement, for it seriously weakens the combustion chamber and interferes with the free flow of the circulating water; consequently cracked combustion heads are fairly common when this design is applied to large engines. From the point of view of governing it has nothing to recommend it, but it provides an inexpensive head, an efficient form of combustion chamber, and a simple valve gear, especially in the case of multi-cylinder engines.

The arrangement shown at B is probably the best from the point of view both of thermal and volumetric efficiency, and is the disposition frequently employed by designers of petrol engines for getting the maximum of power out of a given weight and size of engine. From the governing point of view it is bad, but it does not weaken the combustion head, nor does it interfere with the water circulation to the same extent as the former arrangement. The principal objection to its use is that it is costly, both as regards the combustion head and the valve gear.

The arrangement shown at C is one that is occasionally employed for petrol engines. The inlet valve, being in the cylinder head, can be made of large diameter without weakening the head or interfering with the water circulation, consequently the volumetric efficiency is good, otherwise there is little to be said in favour of this design.

At D is shown an arrangement frequently adopted for both gas and petrol engines. In this case the inlet valve is usually above the exhaust. This disposition compromises between conditions 1 and 2, for, from a thermodynamic point of view, the area of water-cooled surface exposed to the gases is not very great, while it is easy to provide ample valve area and at the same time retain a moderately compact clearance space. If the igniter be fitted in the pocket between the two valves, it is probably the best arrangement for engines which are required to run with widely varying loads, while the valve gear, though not the cheapest and simplest possible design, presents no serious difficulties, and has the additional advantage of being equally suitable for single- and multi-cylinder engines. With this arrangement the valve chamber is integral with the cylinder body, and the combustion head is simply a flat water-cooled plate. In small engines it is not usual to fit a separate head, but to cast the cylinder and head in one piece; in any case the design presents very little difficulty as regards the mechanical features, and does not obstruct the water circulation. In very large

gas-engines, however, considerable trouble is experienced, owing to the difference in the expansion between the inner and outer walls, and cracks are liable to occur in the water-jacket at the point where the jacket wall round the valve pocket joins the main-cylinder water-jacket.

At E is shown by far the most popular arrangement for all petrol engines with both single and multiple cylinders. Its chief claim to favour is its low cost of production, for it is probably the cheapest of all forms, and, from a manufacturing point of view, has many features to recommend it. For instance, it is possible to make the inlet and exhaust valves identical in all respects, and therefore interchangeable; also, by providing detachable covers over the valves through which they can be withdrawn, it becomes unnecessary to use separate seatings and cages. Again, the valve gear is equally suitable for single or multi-cylinder engines. From the point of view of thermal and volumetric efficiency, and also of governing, it is not so good as D, while so far as mechanical strength and the dissipation of heat are concerned it is also inferior to D.

The arrangement shown at F used to be very popular for small multiple-cylinder engines, but it has been almost completely discarded in favour of E, which has most of the advantages of F, and is simpler in every way. This arrangement is, however, still occasionally used for small multi-cylinder petrol engines which are required to run at very high speeds, for it is possible to fit larger valves in this type without lengthening the engine to the extent necessary in the case of E, with all the valves on one side. Thermodynamically, the design is a bad one, for the area exposed in proportion to the volume of the combustion chamber is excessive. It has, however, the merit that the pipework is extremely simple, a point of considerable importance in multi-cylinder engines.

The arrangement shown at G is seldom used in vertical engines, but it has some distinct advantages. It combines little cooling surface and good governing. It is really a modification of D with the valves placed horizontally instead of vertically. As in the case of the arrangement shown at B, the valves are not very easy to operate, and on the whole this disposition is more suitable for horizontal than for vertical engines.

The arrangement shown at H has become practically the standard arrangement for horizontal single-acting engines; and, indeed, it would be hard to find a better disposition, for, although it is not possible to fulfil all the conditions enumerated, it certainly provides

a very fair compromise, and the valve gear as applied to single or multiple-cylinder horizontal engines is exceedingly simple. It is, in fact, probably the ideal valve disposition for single-acting horizontal engines, but it loses much of its charm when applied to vertical cylinders.

The design, both of the valves themselves and of the cams for operating them, is a matter of first importance, especially in very high-speed engines, and here as in many instances one must turn to the modern high-speed petrol engine in order to see what can be accomplished under extreme conditions. It has already been stated that, if the valves be so designed that the velocity through them be about 130 ft. per second, a very good volumetric efficiency can be obtained, and at the same time there will be sufficient turbulence within the cylinder to ensure rapid combustion. The figure, 130 ft. per second, is based on the assumption that the piston is travelling at a uniform speed, and that the valves are fully open throughout the whole of the stroke. From the above considerations it follows that the area of the opening through the valve is solely dependent upon the piston speed, and for a piston speed of, say, 720 ft. per minute, the effective area should be

$$\frac{720}{60 \times 130} \times \text{area of piston} = 9{\cdot}22 \text{ per cent of the area of the piston.}$$

In order to obtain a high volumetric efficiency, it is desirable that the valve shall open and close as rapidly as possible. The actual rate at which it can be opened is, in practice, governed by the spring tension which can be employed. The spring tension again is governed as an upper limit by the space available, and as a lower limit in the case of the exhaust valve by the tension required to keep the valve closed when the pressure in the cylinder is reduced by throttling. In very high-speed petrol engines the upper limit is the controlling one, but in the case of gas or oil-engines, running at a relatively slow speed, this limit is not approached. It is interesting to consider the problem of valve operation in relation to the hypothetical gas or Diesel engine which has been examined previously. This engine runs at a piston speed of 720 ft. per minute, and, as shown above, the effective valve area required to give a mean gas velocity of 130 ft. per second throughout the stroke is 9·22 per cent of the area of the piston, or, in this case, approximately 10·5 sq. in. It may be assumed that the lift of the valve is equal to one-quarter the diameter of the port, so that the

area of opening will be equal to the port area. The diameter of the port will be $\sqrt{\frac{10{\cdot}5}{{\cdot}785}} = 3{\cdot}65$ in., or, allowing for the area of the valve stem, say 3·75 in.; the lift will then have to be approximately 0·9 in. The overall diameter of the valve head will be about $4\frac{1}{4}$ in., and its total reciprocating weight, complete with cap, one half of the valve spring, and such of the actuating mechanism as may be regarded as reciprocating, will amount to about 16 lb. We will assume that the spring tension must be such that when the valve is closed it will resist any possible difference in pressure on either side of it. In the case of the exhaust valve; when the engine is running light under throttle control, this may amount to 10 lb. per square inch, and since the area of the port is 10·5 sq. in., the minimum spring tension required when the valve is closed will be 105 lb., or allowing a reasonable margin, say 120 lb. when the valve is at rest, but as the valve opens the spring is further compressed, and its tension therefore increases. During the first portion of the opening period the valve is forced open by the cam until it has attained its maximum velocity, and the load on the cam during this period is equal to the spring tension, plus acceleration; during the second portion of the opening period the load on the cam is equal to the spring tension minus the acceleration. Similarly as the valve closes the load on the cam is the difference between the spring tension and the acceleration during the first portion of the period and the sum of the two during the latter period. Since it is desirable to open and close the valve as rapidly as possible, the rate of acceleration should be such that the inertia is almost equal to the spring tension at the time when the valve is half open. If the inertia exceeds the spring tension, then it is clear that the valve and actuating mechanism will not follow the cam. The maximum rate of opening and closing is obtained when the inertia, due to acceleration, just balances the spring tension during the latter portion of the opening and the earlier portion of the closing period. It should be noted that the spring becomes a controlling factor only after the valve is half open, consequently if its tension when closed is 120 lb. per square inch the tension when the spring is playing an active part is probably not less than 150 lb., depending upon the normal "rate" of the spring. If we assume that the acceleration is constant throughout the whole period of opening and closing, then it is clear that the rate of acceleration must be such that the inertia of the valve and actuating mechanism does not exceed 150 lb.

The maximum permissible rate of acceleration may be found from the formula,

$$a = \frac{Fg}{W},$$

where a = the acceleration in feet per second per second.
W = the weight of the valve and the reciprocating mass in lb.
F = the spring tension in lb.
g = the acceleration due to gravity, viz. 32·2 ft./sec.^{2}

In this case the equation becomes,

$$a = \frac{150 \times 32 \cdot 2}{16}$$

$$= 301 \cdot 9 \text{ ft./sec.}^2, \text{ or, say } 300 \text{ ft./sec.}^2$$

With an acceleration of 300 ft./sec.^{2} the time taken during one half of the opening or closing period may be found from the formula,

$$t = \sqrt{\frac{2h}{a}},$$

where h = half the total lift in feet = 0·0375.
a = acceleration = 300 ft. sec.^{2}
t = the time in seconds for half the ascent or descent.

In this case,

$$t = \sqrt{\frac{0 \cdot 075}{300}} \quad 50 = 0 \cdot 02 \text{ sec.}$$

The total time of opening or closing is therefore 0·04 sec.

In the case of the engine under consideration the time occupied by each stroke is 0·125 sec. The total period during which either valve is off its seat will, however, be about 220 degrees, and the time of opening will therefore be $\frac{220}{180} \times 0 \cdot 125 = 0 \cdot 153$ sec. Of this period opening and closing take up 0·08 sec., leaving 0·073 sec. during which the valve may remain wide open. Expressed in degrees of crank-angle, 115 degrees of the opening period is taken up in opening and closing the valve, leaving 105 degrees during which the valve is held wide open. This represents the most rapid opening possible assuming

1. That the weight of the valve and such parts of its actuating mechanism as may be considered to comprise the total reciprocating mass amount to 16 lb.

2. That the spring tension when at rest is the lowest compatible with holding the valve shut under the most extreme conditions of throttling, and that its "rate" is such that when half open the tension is increased by 20 per cent.

3. That the acceleration is constant throughout the whole period of opening or closing.

In regard to (3) it is not essential that the acceleration should be constant throughout the whole period. So long as the inertia of the valve is not controlled by the spring, that is to say, during the first portion of the opening period, and the last portion of the closing, a very much higher rate of acceleration may be used if necessary. This affects the loading on the cam only, it does not affect the spring except, in this respect, that under these circumstances the change in acceleration, from positive to negative, takes place earlier in the period when the spring is less than half compressed, and its tension therefore somewhat reduced. Except in extreme cases it is not desirable to operate with a high starting and closing acceleration because, unless the clearance between the valve stem and the rocker beam is very carefully adjusted, the valve is liable to close down on its seating at a fairly high velocity, thus causing noise and undue wear and tear.

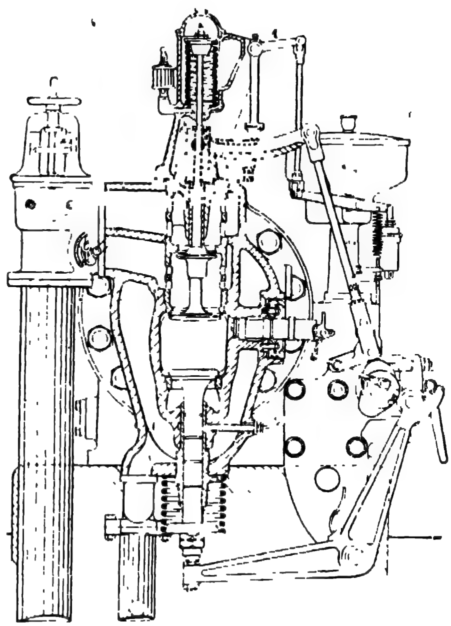


Fig. 38

The customary method of operating both the inlet and exhaust valves in a horizontal gas-engine is shown in fig. 38, which represents a cross-section through the valve chamber of a large Crossley gas-engine. In the example shown governing is effected by varying the lift of the inlet valve instead of by throttling. This is accomplished by introducing a movable fulcrum for the inlet rocker lever, operated by the governor. So long as the valve is at rest on its seating the fulcrum arm can swing freely and without friction, since it is not in contact with the inlet rocker. This is a very neat

and simple method of varying the inlet valve lift, and is certainly most attractive from a mechanical point of view.

In many of the large double-acting engines, and in some of the larger examples of the single-acting type, the valves are operated

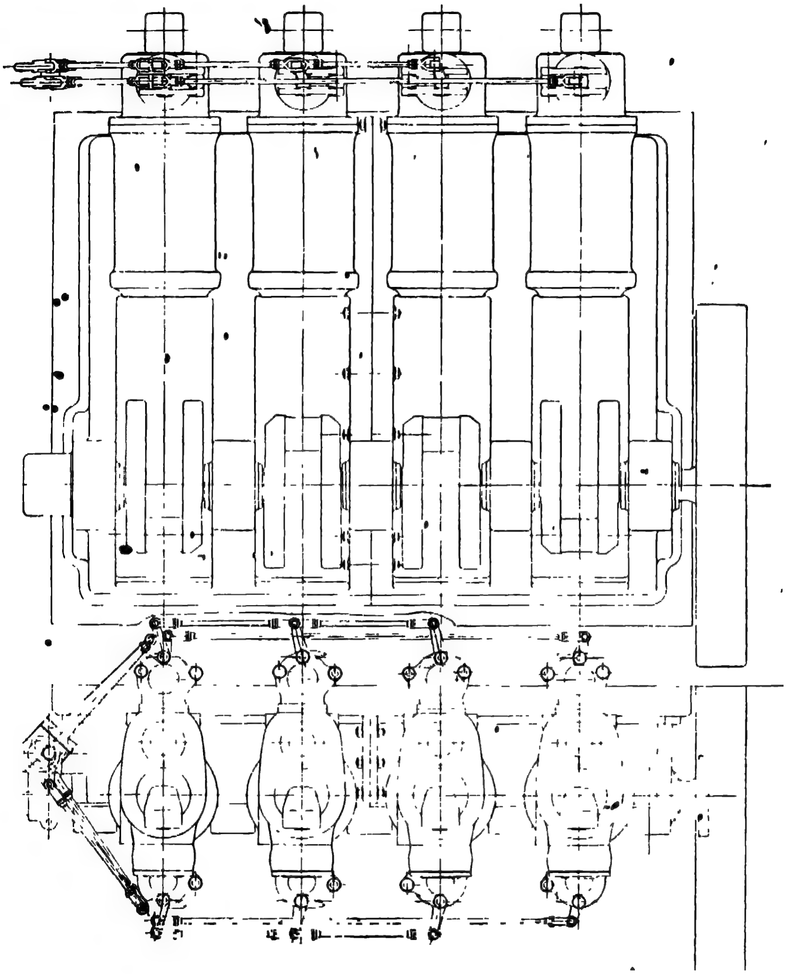


Fig. 39.—Outline Arrangement of 4-cylinder 520-B.H.P. Ruston Proctor Oil Engine

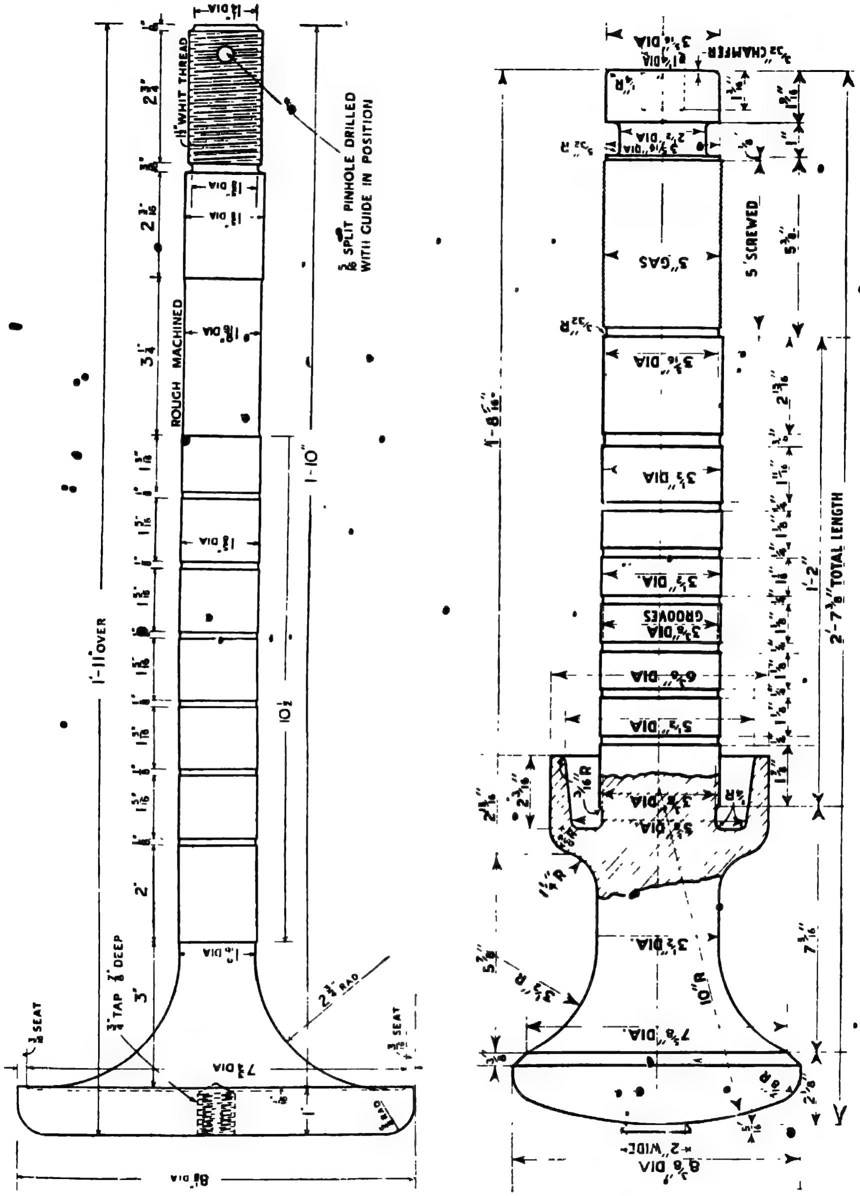
by means of eccentrics on the side shaft. In this case the requisite motion is obtained by the introduction of a system of rolling levers between the eccentric rod and the valve. This method offers great advantages, particularly when the valves are at a considerable distance from the side shaft, since the inertia of the mechanism in both directions is taken care of by the eccentrics. Thus the valve springs are greatly relieved, and have little more than the inertia

of the valves themselves to overcome. By the use of this system, it is possible to operate a very considerable number of valves from a single side shaft and a single pair of eccentrics, as shown in fig. 39, which represents a large 4-cylinder horizontal gas or oil-engine built by Messrs. Ruston Proctor & Co., of Lincoln.

Cooling of Valves.—In the case of the exhaust valve, provision has to be made for dealing with gas at an exceedingly high temperature on both sides of the valve, and means must be provided for withdrawing the heat from the valve head as rapidly as possible. In very small engines the area of the valve is generally so small in relation to its circumference that the heat can be conducted from the centre to the seating without difficulty, and the inlet and exhaust valves can therefore be made identical. In large engines, however, where the area of the valve is great, there is a danger of the centre becoming so hot as to cause premature ignition, also distortion and leakage. To obviate these difficulties, very large exhaust valves are sometimes made hollow, and cooled by water circulation. This is certainly effective, but from a mechanical point of view it is no easy matter to maintain a continual supply of water without leakage. As a general rule, water-cooling of exhaust valves is not adopted, designers preferring to obviate pre-ignition by the use of a lower compression ratio, or a lower mean pressure, even at the expense of a small percentage of thermal efficiency or power. To avoid water-cooling, it is a common practice, in large uncooled exhaust valves, to use very heavy valve heads, and to maintain a tolerably uniform temperature across the head of the valve by employing a great thickness of metal. The heads are usually made of hard cast iron, cast in chills, in order to provide a very hard wearing face, and it is found that this material suffers less than steel from the scoring or erosive effect of highly-heated gases passing over the seating at an exceedingly high velocity. The stems are usually made of nickel steel, and are screwed and riveted into the head. In some cases, however, both the heads and stems are of cast iron. The valves shown in fig. 40 are typical examples of those used in the best modern gas-engines.

In order to withdraw the heat from the exhaust valves as rapidly as possible, the stems are sometimes made of rather large diameter, and are provided with water-cooled guides, which are carried as close up to the head of the valves as possible. One of the principal sources of trouble experienced to a greater or less degree with all exhaust valves is the burning of the stem just under the head of the valve

at the point where the gases impinge upon it. Various methods are adopted to overcome, or at least minimize, this trouble. In very



Inlet Valve

Fig. 40.—Exhaust Valve

large engines it is usual to provide a kind of skirt, made of cast iron, in one piece with the valve head, somewhat as shown in fig. 41; this skirt fits freely over the outside of the valve guide, and so completely protects the stem of the valve from the exhaust products.

The skirt itself, being made of cast iron, does not burn so readily as the steel stem. This arrangement certainly protects the stem admirably, and has much to recommend it, but, since the skirt has to pass outside the guide, and is necessarily of considerable diameter, the effective area of opening is somewhat limited. In smaller slow-running engines, the difficulty is usually met by the simple expedient of using a heavier valve stem, but this, as has already been shown, increases the strain on the junction between the valve head and spindle.

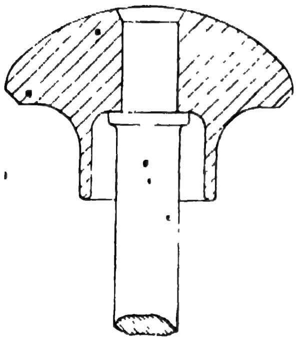


Fig. 41

In small high-speed petrol engines, the problem is very successfully met by the employment of tungsten steel, better known as high-speed tool steel, for the whole valve; this metal retains its tensile strength over a very wide range of temperature and suffers very little from corrosion or burning. Any local weakness is avoided by the provision of an ample radius under the head. Such steel, however, which generally contains about 20 per cent of tungsten and 4 per cent of chromium, is difficult and expensive to machine, and it is doubtful whether it could be successfully applied to larger engines; it has, moreover, the disadvantage that it scales more rapidly than nickel steel. To the engineer accustomed to large slow-running engines, it is amazing to see what these exhaust valves will stand, for they appear to be able to run continuously at an engine speed of over 3000 R.P.M., with not only the head, but a considerable portion of the stem, also at a red heat, and this without distortion or stretching, while the breakage of exhaust valves, even in racing petrol engines, is now a matter of rare occurrence.

CHAPTER XIII

PIPEWORK

Design of Pipework.—It has been stated that if the opening through the inlet valve be such that the gas velocity does not exceed 130 ft. per second, an excellent volumetric efficiency can be obtained, but this is true only provided that the passages and pipework leading up to or from the valve are correctly designed, and also that the valve opens freely into the cylinder. If the valve is placed in a pocket in the cylinder, as is usually the case, care must be taken to ensure that the contour of the walls of the pocket is such that the gases enter the cylinder at a steadily decreasing velocity, and that neither their direction nor their velocity are changed too abruptly. The same thing, of course, applies to the exhaust valve; the gases must be steadily accelerated until their velocity reaches a maximum as they pass through the opening of the valve, and then steadily reduced again. In the design of the valve ports and cylinder, the principle of the venturi tube must be borne in mind, the opening of the valve being regarded as the throat of the venturi. In addition to this, there is yet another point to be considered, namely, the inertia of the gases in the inlet and exhaust pipes, and under certain circumstances this inertia may be made use of to increase the volumetric efficiency.

Taking first the case of a single-cylinder engine of the four-cycle type, which draws in a charge of air or combustible mixture every fourth stroke. At the moment when the inlet valve first opens, the air in the pipe may be assumed to be stagnant, and the piston practically at rest on its inner dead centre. As the piston travels outward on the suction stroke the velocity of the air is accelerated. At about mid-stroke the piston will have attained its maximum velocity, but, owing to the inertia of the air, there will be a certain amount of lag, depending upon the length of the pipe and the weight of air in it, consequently the air will not have attained its maximum velocity until after the piston has passed the

mid-stroke and is slowing up again. When the piston has reached the outer centre and come to rest, the air in the pipe will still be travelling at a high velocity, and will tend to supercharge the cylinder. In other words, during the first portion of the suction stroke, work is done on the air in accelerating it within the pipe, and during the second portion some of the kinetic energy thus generated is converted into static pressure. The timing of the inlet valve should be such that it closes exactly at the moment when the normal flow through the valve would stop. Owing to the lag, this is generally at a point when the piston has travelled through about 40 degrees of the return stroke.

In some exceptional cases, where a very long induction pipe is employed, the lag may be much greater, and it may be advisable to keep the inlet valve open until the piston has travelled through as much as 60 degrees of the return stroke. At the moment when the inlet valve closes, the velocity of the gases in the immediate neighbourhood of the valve has been checked by the increasing static pressure within the cylinder, but the air at the farther end of the pipe still has a considerable velocity, and this in turn tends to pile up a static pressure behind the valve. This continues until the whole of the kinetic energy of the air within the pipe has been so converted; a reversal then takes place, and the air acquires a velocity in the opposite direction. Thus a pulsating effect is set up in the induction pipe. The effect of this first reversal is to produce a blow-back at the mouth of the pipe. Such a blow-back, however, does not necessarily mean that the inlet valve is leaking or that the timing is incorrect; it is, in fact, a normal and healthy condition.

Now, if the pulsations set up in the induction pipe synchronize with the period of opening of the inlet valve, so that the air in the pipe has an initial velocity in the direction of the cylinder at the commencement of each suction stroke, it follows that the inertia effect is greater, and that the pulsations will increase with each suction stroke so long as they remain in synchronism or until they have reached a maximum. If this be the case, the quantity of air taken into the cylinder will be considerably increased; but if, on the other hand, the pulsations are out of phase with the period of the inlet valve opening, and the air has acquired a velocity in a direction away from the cylinder at the moment when the valve closes, a smaller charge will be taken into the cylinder at each stroke until the pulsations have died down again.

From the above considerations it is clear that if a long induction pipe be fitted, the volumetric efficiency, and therefore, of course, the mean effective pressure, will vary according to whether the periodicity, or some function of the periodicity, of the pulsations of the induction pipe synchronizes with the period of the inlet valve. When these pulsations are in phase the volumetric efficiency will be above, and when they are out of phase it will be below the normal. As a general rule, long induction pipes are to be avoided, because, even though the average result over a long period may be unaffected, they are liable to produce periodic changes in the mean effective pressure, which are detrimental to the governing of the engine. In engines which are always required to run at a uniform speed, and in which the charge of air is always uniform, it is conceivable that the length of the induction pipe might be adjusted so that the pulsations were always in synchronism with those of the inlet valve, and an increase in volumetric efficiency and power might be gained thereby. But in engines in which either the speed or the weight of charge, or both, are varied, it is not easy to see how advantage could be taken of these pulsations, unless the induction pipe could be made in the form of a trombone, which is hardly practicable.

A somewhat similar effect is produced in the exhaust pipe, but here the conditions are not quite the same. When the exhaust valve is first opened, there is generally a considerable pressure in the cylinder, and the velocity of the gases is very high indeed. This immediately produces a high velocity in the exhaust pipe, so that the maximum velocity is reached long before the piston has attained its highest velocity. In consequence, in this case, instead of a "lag", there is a substantial "lead" in the gas velocity over that of the piston, and if there be no serious resistance at the end of the exhaust pipe, the inertia of the gases will tend to reduce the pressure in the cylinder to below atmospheric. When the exhaust valve is closed, the same reversal will take place, and the same pulsations will be set up; consequently the pressure in the cylinder at the end of the exhaust will depend to some extent upon whether the periodicity of the pulsations is in or out of phase with the exhaust-valve opening. In this case, however, the initial velocity is so much greater that the influence of the pulsations is much reduced.

In practice, the use of an efficient muffler at the end of the exhaust pipe generally produces so serious a resistance to the free flow of the

gases that full advantage cannot be taken of their inertia. The effect, however, is clearly shown in the indicator diagrams (fig. 42), in which the exhaust line following an explosion is much lower, and shows less back pressure than when no explosion has taken place, and the gases have no high initial velocity. These are actual indicator diagrams taken from the author's experimental engine, to which a fairly long and tortuous exhaust pipe was fitted, and in which the muffler was too small and offered serious resistance to the free flow of the gases. Assuming that the pressure of the gases in the cylinder amounts to 45 lb. per square inch absolute, at the moment when the exhaust valve is first opened, and that their weight, taking into account their high temperature, is 0·02 lb. per cubic foot, then, if there be no friction, their initial velocity may be found from the equation:

$$V^2 = \frac{2g(P - p)}{W},$$

where V = the velocity through the exhaust valve,
P = the initial absolute pressure,
p = the final absolute pressure,
W = the weight per cubic foot.

In this case

$$V^2 = \frac{64 \cdot 4 \times 144\ (45 - 15)}{0 \cdot 02},$$

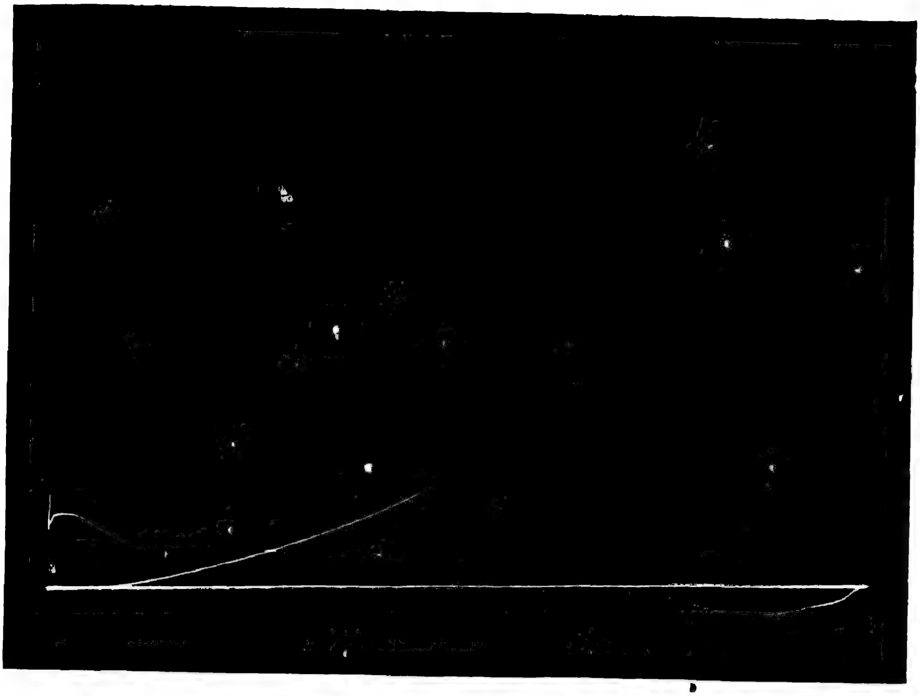
$$V^2 = \frac{64 \cdot 4 \times 144 \times 30}{0 \cdot 02},$$

$$V^2 = 13900000,$$

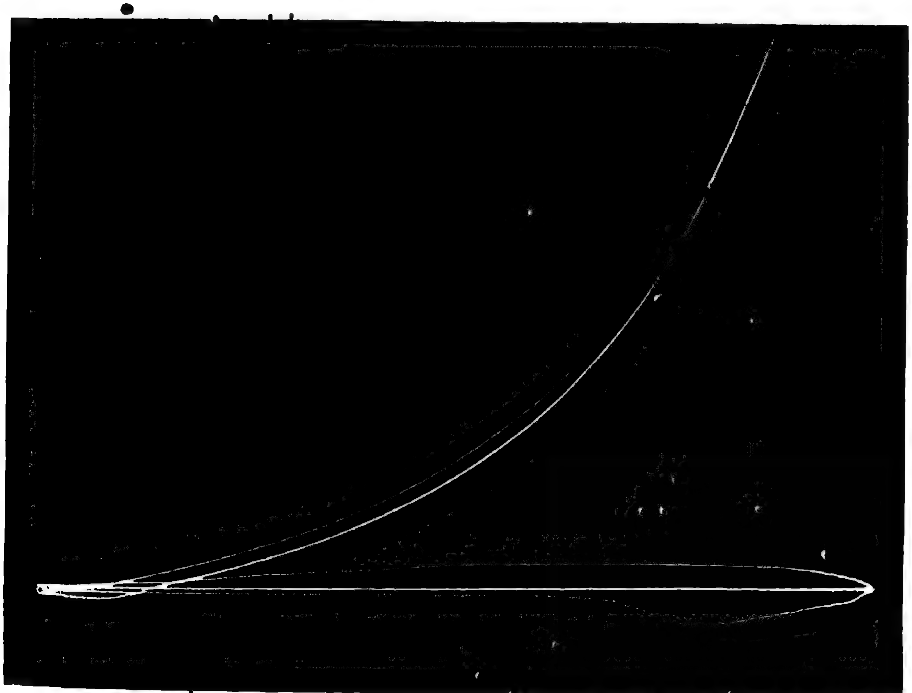
$$\text{and } V = 3715 \text{ ft. per second.}$$

This is the velocity through the valve port at the moment when the exhaust valve is first opened, assuming no friction loss. The velocity in the pipe will be very much lower, because the area of the pipe should be at least 50 per cent greater than the maximum area of the valve; moreover, the gases are cooling and contracting rapidly. But taking all this into account, the maximum velocity in the exhaust pipe may still be well over 1000 ft. per second, a figure far in excess of anything obtained in the inlet pipe.

To charge the cylinder fully, one has only the pressure of the atmosphere to rely upon; to empty it, one has both the high initial velocity in the exhaust pipe and the positive action of the piston in one's favour. From all the above considerations it is fairly clear



(*a*) Full Load. Indicator Spring 20 lb. = 1 inch



(*b*) Not Firing. Indicator Spring 20 lb. = 1 inch

Fig. 42

that attention must be concentrated on the inlet valves and inlet piping rather than on the exhaust.

Piping of Multi-cylinder Engines.—So far only single-cylindered engines have been considered, but when two or more cylinders are employed the problem becomes more complicated. In most large multi-cylinder horizontal engines separate and entirely independent induction and exhaust pipes are fitted to each cylinder, and the problem is thus the same as in the case of a single-cylinder engine; but in many high-speed vertical gas-engines and petrol engines it is customary for all the cylinders to draw from a common carburettor or throttle chamber, and to exhaust into one common pipe or manifold. In such cases the induction and exhaust piping must be designed with the greatest care.

Taking the case of two-cylinder engines first. If the cranks are set at 360°, and both pistons travel together, the expansion and suction strokes occur at equal intervals, and the effect is precisely the same as in a single-cylinder engine running at double the number of revolutions. But when the cranks are set at 180°, as is commonly the case in high-speed engines, two suction strokes follow each other successively, and are again followed by two exhaust strokes. In this case No. 1 cylinder first draws from the induction pipe, and creates a high velocity therein, but the full advantage of this is not realized in No. 1 cylinder, because, long before the gases can come to rest, the inlet valve of No. 2 cylinder is opened, and their velocity is again accelerated by No. 2 piston. At the end of this stroke full advantage is taken of the high velocity and kinetic energy of the gases to raise the static pressure in No. 2 cylinder; consequently this cylinder receives a larger charge. The same thing applies to the exhaust if a fairly long pipe be used, for the opening of No. 2 exhaust valve may tend still further to accelerate the velocity in the common exhaust pipe. If the flow of the exhaust gases be restricted, then the sudden rise of pressure, which occurs when No. 2 exhaust valve is opened, may drive them back into No. 1 cylinder. On the other hand, if the pipe be long and the flow quite free, the sudden increase of velocity in the pipe may create a partial vacuum in No. 1 cylinder. In either case the result will be irregular, and the positive or negative pressure in the two cylinders at the end of their exhaust strokes will not be the same.

In three-cylinder engines the cranks are always set at 120°, and the intervals between the suction strokes are equal, but the cylinders draw in their charge in the order 1, 2, 3, 1, 2, 3. It is a common

practice to employ a single straight induction pipe running alongside all three cylinders. In this case the gases are steadily accelerated along the pipe and reach a maximum velocity while cylinder No. 3 is taking in its charge, consequently this cylinder receives the largest weight of charge, while in order to supply No. 1 cylinder the flow of gases in the pipe must be brought to rest and reversed, with the result that No. 1 cylinder is starved. In the case of a gas-engine or Diesel engine, which do not rely upon the velocity in the inlet pipe to retain the fuel in suspension, this difficulty can be largely overcome by the employment of an induction pipe of very large diameter, so that the velocity is exceedingly low and the

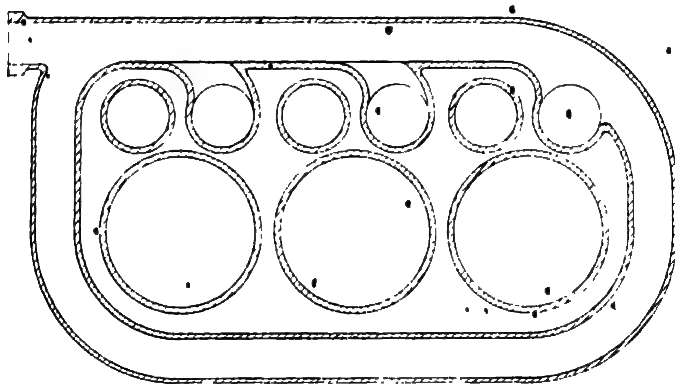


Fig. 43

inertia forces are not serious; but in the gas-engine the common induction pipe should be divided by a diaphragm, in order to separate the gas from the air, which should only be allowed to mix as they pass through the inlet valve of each cylinder. Otherwise, in the event of a back-fire through the inlet valve, due to a slow-burning mixture, on the previous cycle, the whole of the combustible mixture in the large common induction pipe will be ignited, and for the next few cycles the engine will draw in products of combustion instead of combustible mixture.

In any engine employing petrol or vaporized paraffin a large-diameter pipe cannot be used, because, unless the gases be maintained at a high velocity, the vapour may condense, and any unvaporized particles of fuel will be precipitated on the walls of the pipe. Hence means must be found for providing equal distribution to all cylinders, and at the same time for maintaining a high velocity. The simplest method is probably to use three separate pipes, one to each cylinder leading from the common carburettor or vaporizer.

In this case the length of each pipe should be as nearly equal as possible in order to ensure that the periodicity of the pulsations in each shall be the same. Another method, and one which has been employed by the author with some success, is illustrated diagrammatically in fig. 43. In this case a single induction pipe is employed with three short branches, one leading to each cylinder, but the pipe is carried completely round the three cylinders, in order to form a closed circuit and allow the gases to flow continuously in one direction. The supply from the carburettor or throttle valve is then fed into this pipe tangentially. This arrangement permits of a high velocity being maintained continuously in the induction pipe, and provides for equal distribution to all cylinders.

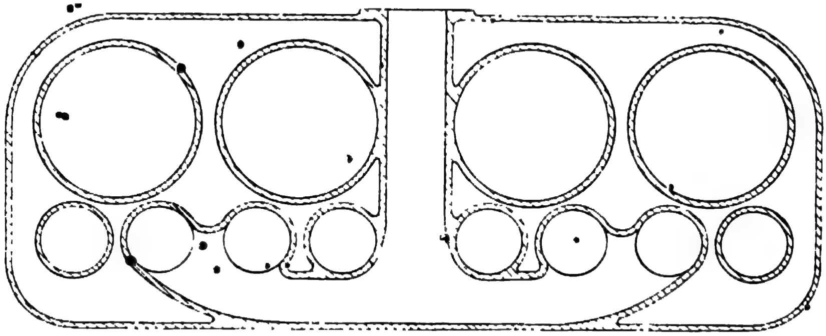


Fig. 44

Piping for Four-cylinder Engines.—In four-cylinder four-cycle engines the cranks are set so that the two outside and the two inside pistons respectively rise and fall simultaneously. That is to say, when the two inside pistons are on the bottom dead centre the two outside are on the top centre. In this case the order of firing, and therefore of charging, is 1, 3, 4, 2. This may be also written 3, 4, 2, 1; 1 and 4 being the outside cylinders, and 2 and 3 the inside cylinders. From this it will be seen that pair numbers 3 and 4 draw their supply of working fluid consecutively from the inside outwards, and are followed by pair numbers 1 and 2, which also draw their supply from the inside outwards; thus the direction of flow must be reversed between 4 and 2, and between 1 and 3, or twice per cycle.

To obviate this all sorts of arrangements have been adopted, and that usually employed is shown in fig. 44, in which the carburettor is shown at A. From A to B a single pipe, or cored passage, is led through between the central pair of cylinders, and then

branches into two, leading to the inlet valves of the two pairs of cylinders. These valves are placed side by side, with the exhaust valves on the outside of each pair. In the pipe A B no reversal takes place, and the gases flow continuously from A to B, but in the branch pipes B C and B D reversals do take place, the gases flowing alternately in the direction B C and B D. These, however, can be kept very short, so that the inertia forces of the gases are not of any consequence.

A second arrangement, and one that is very frequently employed, particularly when cylinders are cast separately, is illustrated in fig. 45. In this case short connecting pipes serve to connect the inlet valves of each pair of cylinders together, and each pipe is supplied by a separate induction pipe leading from the carburettor. This does not appear to be a very good arrangement, because in each pair of cylinders there are two successive suction strokes followed by two strokes during which the inlet valves are closed; consequently the flow of the gases in each induction pipe is arrested alternately, and pulsations set up. Moreover, in each pair the second cylinder will receive a larger charge than the first, since, as in the case of the two-cylinder engine, the second cylinder takes advantage of the high velocity created in the induction pipe by the first cylinder. It will be noticed that in this case there is no

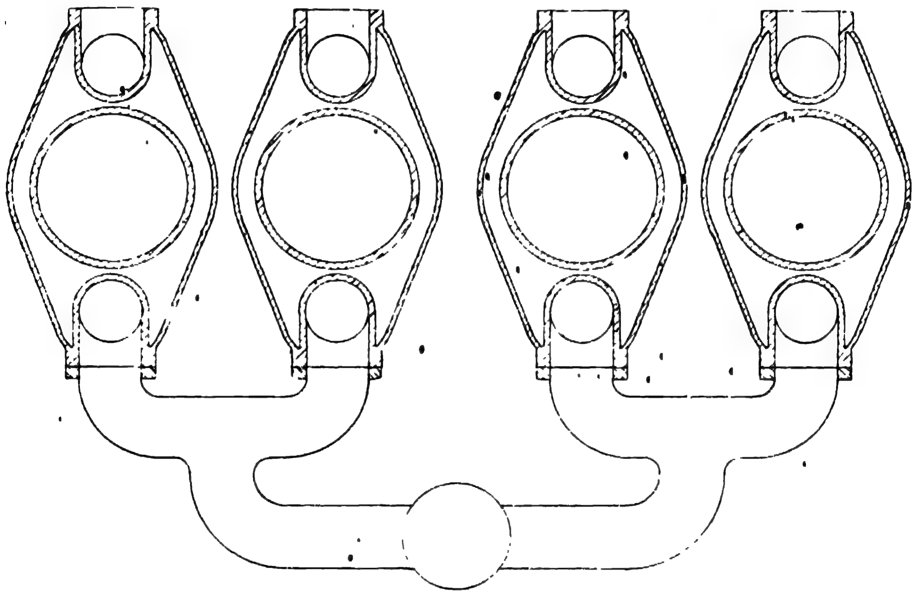


Fig. 45

part of the inlet pipe in which the flow is continuous and in one direction.

The arrangement shown in fig. 46 is one that is sometimes adopted for very high-speed engines, in which it is desired to obtain the maximum possible horse-power from a given size of cylinder. In this case a closed circuit is employed for each pair of cylinders, and, as can be readily seen from the diagrammatic illustration, the flow of the gases is continuous and in one direction. This arrangement, however, is bulky, and if used for petrol or paraffin it presents a very large amount of surface for condensation, which must be

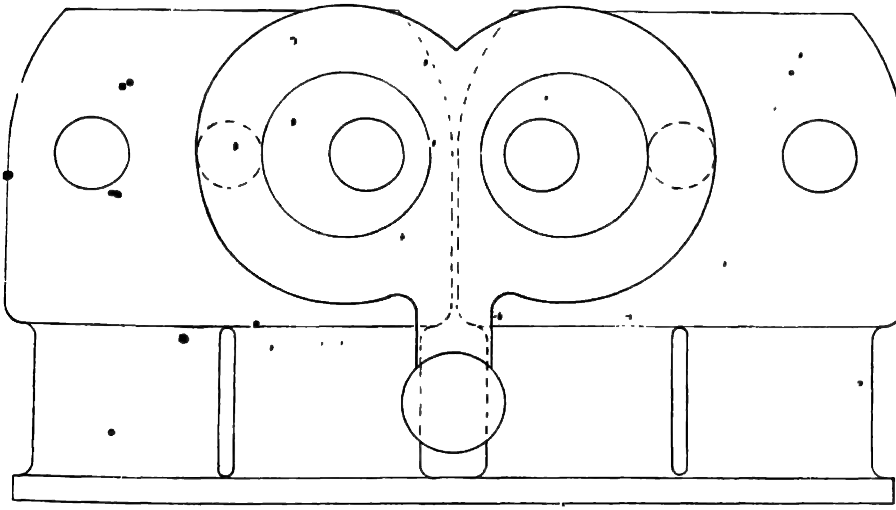


Fig. 46

serious when the engine is throttled down and the velocity is much reduced. Since each pair of cylinders is supplied independently, it is clear that the same arrangement is equally applicable to two-cylinder engines, with cranks at 180°, though the author has never seen it applied to such engines.

Piping for Six-cylinder Engines.—In the case of six-cylinder engines the problem is still more complicated, and since the arrangement and relative angles of the cranks are determined by the conditions of even-turning moment and balance there is very little choice in the order of firing, the usual order being 1, 4, 2, 6, 3, 5. A mere glance at this sequence suffices to show that no closed circuit system of piping is possible, nor can reversals or pulsations be avoided; also, owing to the great length of the engine the induction piping must necessarily be very long, and therefore the

magnitude of the inertia forces becomes serious at high speeds. For this reason nearly all six-cylinder engines are provided with two or more carburettors, or throttle chambers, when it is required to obtain the best possible power or efficiency from them, as, for instance, when they are employed for aeroplanes. As a general rule six-cylinder engines are used only for motor-cars, aeroplanes, and marine work. In the former case the engine usually has so large a margin of power that a poor volumetric efficiency can be tolerated, and a single carburettor, with a system of pipework involving the minimum possible amount of interference, is adopted. In the case of large marine oil-engines, in which air only is taken into the cylinders, no such difficulty exists, because each cylinder can be fitted with a separate short length of pipe open to the atmosphere.

Exhaust Piping.—Taking next the question of exhaust piping. The design of this must depend both on the timing of the valves and on the type of muffler, or conversely the timing of the valves must depend upon the arrangement of exhaust pipe. In single-cylinder engines if a long exhaust pipe can be provided, free from sharp bends and opening into a muffler of large capacity, which offers little resistance, then advantage can be taken of the inertia of the gases in this pipe for scavenging the combustion chamber, and the inlet and exhaust valves can "overlap". If, however, owing to the design of the engine-house, there is not space for a large muffler, or if for any other reason a pipe of sufficient length cannot be advantageously used, then half measures should be avoided and the muffler should be placed close up to the cylinder. In such a case no overlap should be allowed in the timing of the valves. As a general rule, it may be said that it is always preferable to employ separate exhaust pipes to each cylinder whenever possible, and in no case should these be led into one common receiver. If a common receiver is inevitable, as is usually the case on motor-cars, then the receiver should be kept as far as possible from the cylinders and the separate pipes made to project well into it, or deflectors used to prevent the gases from one cylinder reacting back into another. Nothing can be worse than the arrangement only too often seen in manufacturers' test rooms, where a large common exhaust pipe is provided, into which a number of engines exhaust through short branch pipes. Such an arrangement is liable to produce disturbances of such magnitude as to interfere seriously with the general running of all the engines, and more especially is this the case when the

engines operate on the two-stroke cycle, a cycle which is very susceptible to disturbances in the exhaust. The pulsations in one long common exhaust pipe may, under conditions of synchronism, reach a magnitude which few people realize.

In the four-cylinder engines used for motor-cars and marine work, the use of four separate exhaust pipes is sometimes impossible, and in these cases all four cylinders exhaust into one common manifold, fitted close up to the cylinders, and in some cases cored into the same casting. When a single manifold is employed, the branch pipes from the cylinders leading into it should be of com-

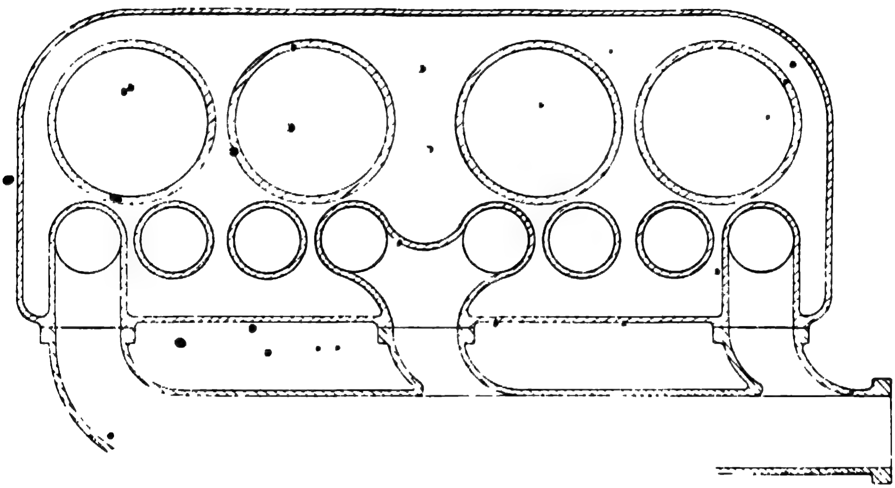


Fig 47

paratively small bore, and should meet the manifold tangentially, so that the exhaust gases from all four cylinders enter it at a high velocity, and all in the same direction. Unless this is done there will be a considerable amount of blow-back of the exhaust gases from the cylinder which has just started to exhaust, into that which is just completing its exhaust stroke. Attention must be paid to the sequence of firing, and the cylinders should be connected to the manifold as shown in fig. 47, and not in any case as shown in fig. 48, an arrangement sometimes adopted. In the arrangement shown in fig. 48, since cylinders 3 and 4 exhaust successively, and their exhaust valves are open for about 220° of the crank in each case, it follows that there is a period during which both valves are open simultaneously, and the exhaust gases issuing from No. 3 cylinder at a high pressure, will enter No. 4 cylinder just before the completion of the stroke, and thus introduce a large quantity of exhaust pro-

ducts immediately before the commencement of the suction stroke. This will have a very deleterious effect upon the volumetric efficiency of that cylinder. It is often found convenient to arrange the exhaust valves of Nos. 2 and 3 so that they both exhaust into one common chamber immediately behind the valves, and since the exhaust strokes in these two cylinders take place a whole revolution apart, there is not the least fear of overlap. There is therefore no objection to this arrangement, which is neat and convenient.

In the case of very high-speed four- or six-cylinder engines, if the exhaust is perfectly free, as in a racing motor-car or aeroplane

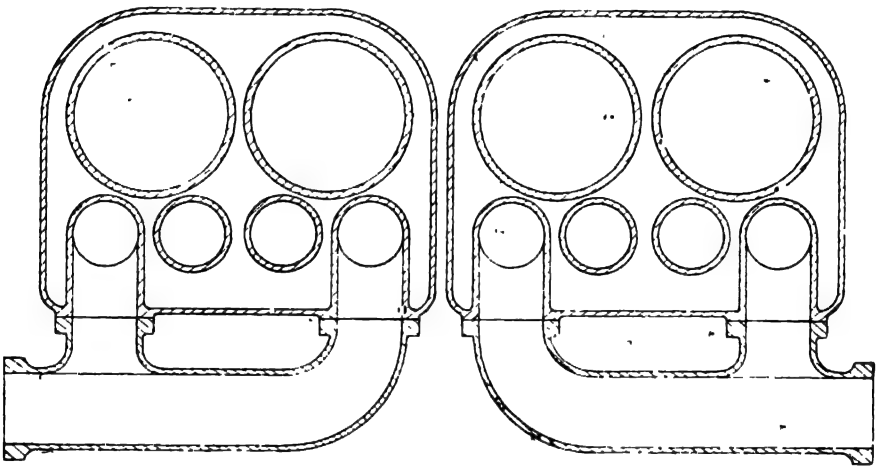


Fig. 48

engine, a considerable advantage will be gained by using four or six separate exhaust pipes for some distance from each cylinder, and then merging them all into one long common pipe. Since the exhausts from four or more cylinders overlap, a continuous high velocity can be maintained in the common pipe, and advantage taken of this to produce a partial vacuum in each cylinder at the end of the exhaust stroke, or, if the valves be timed to overlap, to provide a certain amount of scavenging. Such an arrangement is only of advantage when a perfectly free exhaust can be permitted, but owing to the terrific noise it cannot usually be tolerated.

From the above considerations it will readily be understood that the design of the pipework is a matter that deserves a great deal of consideration. In multiple-cylinder engines, efforts should be made to obtain a continuous flow in one direction throughout the inlet and exhaust pipes, so that the changes in velocity may be utilized

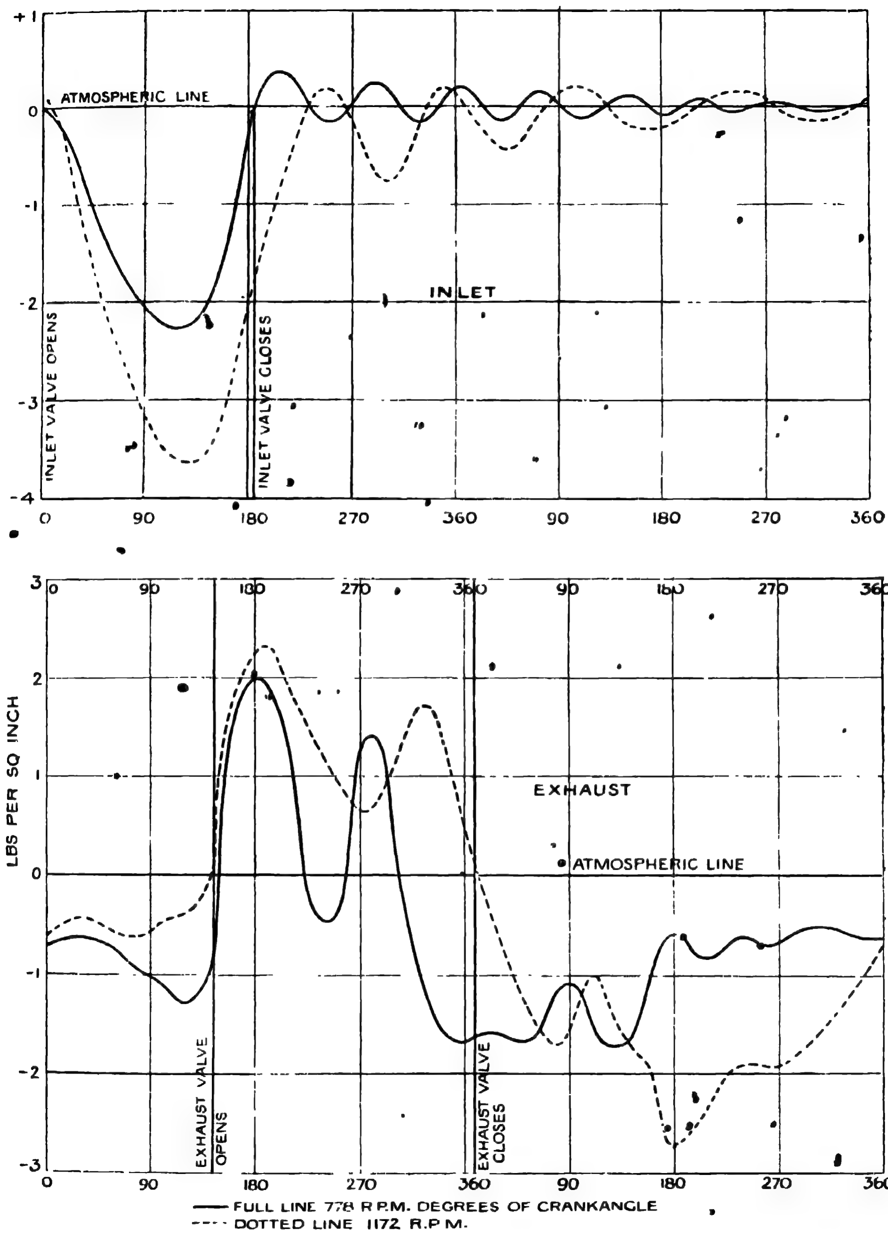


Fig. 49

to obtain a partial vacuum in the cylinders at the end of the exhaust, and a slight supercharge at the end of the suction stroke.

Where it is not possible to obtain a continuous flow in one

direction, the designer must decide whether he will attempt to make use of the reversals and pulsations that occur or not. If he considers that the conditions under which his engine will be running are such that the inertia forces set up by these pulsations can be made to serve a useful purpose, he should arrange his pipework accordingly, using fairly long pipes of comparatively small bore, and he should be careful to ensure that the timing of opening and closing of his valves is correctly adjusted to suit the magnitude and periodicity of the pulsations. To ascertain if this is the case, an optical indicator should always be employed, for the inertia of the parts of a pencil indicator is so great in relation to the very light spring that must necessarily be used, that the results obtained cannot be relied upon. The indicator diagrams, illustrated in figs. 47, 48, are taken from the induction and exhaust pipes of a four-cylinder high-speed engine, and serve to show the kind of diagram that can be obtained, and the effect of the inertia of the gases in the pipes. When, as is more frequently the case, the designer decides that he cannot conveniently make use of the pulsations and inertia forces, he must take care to ensure that they are reduced to a minimum by the employment of short and large-diameter pipework. In no case should the pulsations be ignored, for their effect may be much more serious than is commonly supposed; also, in no case should a number of independent engines be arranged to draw their supply from a common ring main, unless separate receivers of large capacity are fitted close to each engine, nor should they be allowed to exhaust into a common exhaust pipe.

The whole question of pipework is a matter of extreme importance, and it is no great exaggeration to say that in the design of an engine the pipework should first be arranged, and the rest of the engine put in where there is room for it.

From all the above considerations it is clear that no definite rule can possibly be given as to the timing of the inlet and exhaust valves, for this must necessarily depend upon a great number of different factors.

CHAPTER XIV

MECHANICAL DETAILS

Lubrication.—The system of lubrication to be employed must depend both upon the size and speed of the engine. For large comparatively slow-running engines it is usual to lubricate the main crankshaft bearings by means of ring oilers, as shown in fig. 50; that is to say, slots are cut in the upper halves of the main bearings at right angles to the centre line of the shaft, so that a portion of the crankshaft is exposed. Plain rings, generally about 60 per cent larger in diameter than the crankshaft, are threaded over it, so that they rest upon the shaft in the slots, and are kept in position by them. The lower parts of the rings dip into troughs below the bearings, which are kept well filled with oil. As the shaft revolves these rings also revolve,

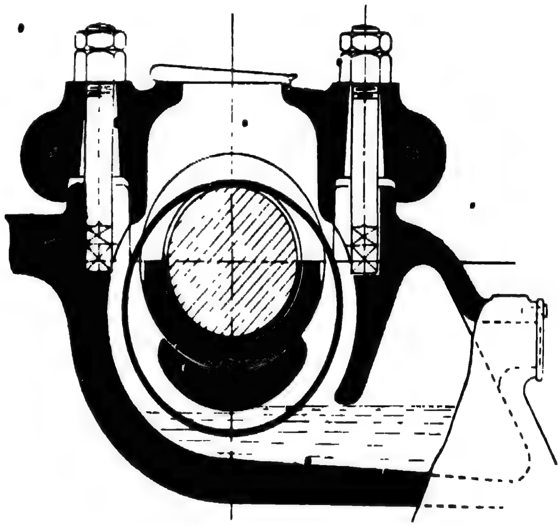


Fig. 50

but at a slower speed, due partly to their greater diameter and partly to the retarding effect of the viscosity of the oil. In this way the oil is continually being dredged by the rings, and delivered to the crankshaft and bearings through the slots on the top. For medium and slow-speed engines, where the bearing pressures are comparatively low, this system works admirably, for it is thoroughly reliable and easy of inspection.

For the crankpin bearing, what is commonly known as a banjo lubricator, shown in fig. 51, is generally fitted; this consists of a concentric brass trough attached to the side of the crankwebs, into

which oil is fed by a sight-feed lubricator. From the trough, holes are drilled through the crankwebs and crankpin, and the oil is forced through these to the connecting-rod big end bearing by centrifugal force. The outlet hole on the crankpin should preferably be drilled in such a position that the oil is not delivered at the point of maximum pressure; that is to say, it should be drilled at an angle of about 45° to the sides of the crankwebs.

The lubrication of the pistons of single-acting engines is usually provided for by means of a sight-feed lubricator working in conjunction with either a dredger or small plunger pump driven from the camshaft, and arranged in the case of horizontal engines to deliver oil to the top side of the piston; suitable ducts or pipes being fitted to the piston to convey some of this oil to the connecting-rod small-end bearing. For vertical engines of the open type, and also for double-acting engines, the oil is forced into the cylinders by means of plunger pumps, and delivered at three or four points equidistant around the circumference, in the same manner as is usually adopted in steam-engine practice, when using superheated steam.

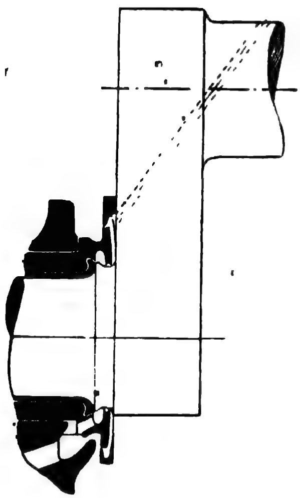


Fig. 51

For all engines running at very high rotative speeds forced lubrication should be employed for the main bearings, and the oil should be supplied at a pressure of from 5 to 50 lb. per square inch. In this case the same oil is circulated over and over again, and collected in a trough or sump at the bottom of the base chamber. Before being circulated a second time the oil should be made to pass through a filter of fine wire gauze, and also, in the case of large or severely loaded engines, through an oil-cooler of ample cooling surface; the provision of such an oil-cooler makes for more efficient lubrication and for economy of oil. When forced lubrication is employed the engine must, of course, be totally enclosed, and provision should be made for ventilating the crank-chamber, otherwise, owing to the high temperature of the pistons, some of the lubricating oil may be vaporized and an explosive mixture formed therein. Several serious accidents have already occurred from this cause. Further details of the methods of lubrication employed in different types of engines will be given when dealing with each particular type in detail.

Bearings.—As a general rule, for all main and connecting-rod bearings white-metal linings should be employed, and phosphor-bronze bearings should seldom be used except in conjunction with case-hardened steel journals. The white metal should be cast into a steel, or cast-iron, or better still, a bronze shell. In any case the surface of the shell should always be well tinned before the white metal is run in; this will ensure good contact and reduce the risk of the white metal breaking away. When cast-steel or cast-iron shells are used, a number of dovetailed grooves should be turned or cast, so that the white-metal lining is keyed in position. It is also advisable to hammer the metal well after cooling, in order to stretch it and so take up the contraction which occurs on cooling. It is most important that the white metal should make a good metallic contact with the shell, for, not only is it liable to break away if not uniformly supported, but there is also danger of a certain amount of lubricating oil finding its way between the white metal and its shell. Since oil is a poor conductor of heat, this may lead to overheating and fusion of the bearing metal. In very small engines it is becoming customary to employ die-cast white-metal bearings without any separate shell. These bearings are cast dead to size in metal moulds, and require no machining beyond a little scraping on the bearing surfaces. The practice, however, is not to be recommended, because oil can find its way behind the white metal, and so insulate it and prevent it from conducting its heat away. An excellent method which has lately come into vogue for small bearings is to cast the white metal into the shell which has been previously tinned, and then drive through it a taper mandril, the larger end of which is exactly equal to the diameter of the shaft. By this means the metal is thoroughly compressed and bedded and an excellent wearing surface produced. For the small end of the connecting-rod it is usual to employ phosphor-bronze bearings, working in conjunction with a case-hardened steel gudgeon-pin. For small engines this answers admirably, and these bearings in practice give little trouble and show but little wear in spite of the enormous loads and high temperatures to which they are subjected.

Crankshaft.—The design of the crankshaft does not call for very much comment. Formulæ for calculating the strength and size of every part of a crankshaft are to be found in nearly every textbook, but the actual dimensions are generally fixed, not by the strength required, but by such considerations as bearing surface and rigidity. That opinions differ very widely on this subject can easily

be gleaned from the fact that different makers use crankshafts of which the diameters vary by as much as 40 per cent for engines of the same leading dimensions. For explosion engines it is perfectly safe to reckon upon a maximum pressure of 700 lb. per square inch under any conditions, except that of a seized piston; but for Diesel-type engines, as has already been explained, much higher pressures may have to be dealt with under abnormal conditions. Generally speaking, if the crankshaft is large enough in diameter to be free from whip, and to provide the bearing surface necessary, it will have more than sufficient strength. To provide adequate bearing surface the projected area of the crankpin should, as a general rule, be at least 30 per cent of the area of the piston for a piston speed of 1000 ft. per minute, and, unless an unduly long crankpin be employed, the diameter of the pin will be equal to nearly half that of the piston, which is considerably greater than is needed to resist any bending stress which may occur. Much, of course, must depend upon both the speed and the method of lubrication, also upon whether the bearings are well ventilated or situated in an enclosed crank-chamber. In the case of multiple-cylinder engines, it is advisable, in order to avoid torsional vibrations, to employ still heavier crankshafts. Since crankshafts are usually machined from a flat slab of metal, the depth of the webs is generally reduced to a minimum, in order to use the thinnest possible slab. This means that, to obtain the necessary stiffness, they have to be made somewhat wider than would otherwise be necessary; this increases the distance between the bearings, and therefore the bending moment.

Connecting-rod.—This does not call for much comment. It is clearly important that the small or piston end of it shall be kept as light as possible, for this is reciprocating weight. It is usual in gas-engine practice, and indeed in all except petrol-engine practice, to employ adjustable bearings for the gudgeon-pin end of the rod. It is very much open to question whether this practice is desirable, except in large engines, for it involves a very considerable addition to the weight of the reciprocating parts, and the adjustment provided is not of very much value in small engines, for, when once the gudgeon-pin has been withdrawn, it is quicker to drive out the bush and to fit a new one than to adjust and scrape in the split bearing. If the gudgeon-pin be of case-hardened mild steel, and the bush of hard phosphor-bronze, the amount of wear that takes place is exceedingly small. The connecting-rod itself is almost invariably

machined from a steel forging, though in petrol engines it is now customary to use stampings in high tensile steel. From the point of view of weight and rigidity there is little doubt that the I section is the best; but, except in small rods that can be stamped, this section is a very expensive one, since it can only be obtained by milling out from a rectangular forging. For this reason nearly all the connecting-rods in use in gas and oil engines are circular in section, this being of course the least expensive form. In all types of internal-combustion engines the bolts of the big-end bearings are always a source of weakness. Owing to the severe reversals of stress a certain amount of deformation of the big-end bearing generally takes place, and in consequence severe stresses are thrown upon the bolts; moreover, the continual reversal of stress tends to cause the metal to crystallize and become brittle; for this reason it is generally advisable to renew, or, at least, to anneal the big-end bolts from time to time. In most of the best class of engines it is customary to forge the big-end bearing cap in one piece with the rod, and afterwards part it off through the centre line. This undoubtedly reduces the tendency to deformation, and therefore the stresses on the bolts, but it is an expensive construction. Another method frequently adopted is to use either stamped or cast-steel bearing caps, which is certainly a cheaper form of construction. The ordinary marine type of bearing, though frequently employed, has not been found to be altogether satisfactory, and there have been a great many accidents due to broken big-end bolts with this type of bearing, which necessitates the use of very long bolts. For the lining of the big-end bearing ordinary white metal is now invariably employed, and has been found to be entirely satisfactory, provided that it is properly supported by its shell and cannot break away or "flow". During the last few years the use of cast-steel connecting-rods has come into vogue. This practice hails from America, and has much to recommend it on the grounds of low cost of manufacture. As in the case of stampings, cast-steel connecting-rods can be made of I section without increasing the cost, and this section is therefore almost invariably employed. For high-class engines the use of cast steel is hardly to be recommended, for although enormous strides have been made in steel foundry work during the last few years, this material is hardly yet as reliable as forged or stamped steel.

Cylinder.—In all but the very smallest horizontal, and in nearly all the larger sizes of vertical engines, separate liners are employed in the cylinders. These liners are made of a very hard

quality of cast iron, generally containing a large proportion of manganese, or, in some cases, steel scrap. They are, or should be, machined all over inside and out, and should be kept as thin as possible so that the temperature gradient through the metal shall be reduced to a minimum. In horizontal engines it is usual to bolt the liner securely to the combustion head, leaving the other end free to expand longitudinally. The water joint at the outer end is generally made by means of a rubber ring inserted in a groove turned round the outside of the liner. The thickness of the rubber is such that it projects well above the top of the groove until it is compressed into position in the cylinder-jacket casting. This arrangement is perfectly satisfactory, and trouble from leakage between the cylinder jacket and the liner is almost entirely unknown. In vertical engines it is usual to increase the thickness of the top few inches of the liner and press it into the cylinder jacket, which is bored out to receive it. The bottom joint is almost invariably made water-tight by means of a rubber ring as in horizontal engines, but in a few cases, as, for example, in the large Carels-Diesel engine, the rubber ring is dispensed with and the liner made a good sliding fit in the jacket casting, leakage being prevented by means of red lead. The principal advantages in favour of the use of separate liners are:—

1. The liner, being an extremely simple casting, can be made of a material which will give a good hard wearing surface.

2. Being circular in section, and of practically uniform thickness throughout, it will not distort when heated.

3. Being securely held at one end only it is free to expand longitudinally without any restraint.

4. In case of wear it can be renewed at a comparatively small cost.

5. Being machined all over inside and out, and being of harder material, it can be made thinner than would be the case if it were made of the same material as the cylinder jacket, for, since this latter is almost invariably a somewhat complicated casting, a metal must be chosen which will flow freely in the mould and will withstand the stresses due to contraction. Such a metal does not usually give a good wearing surface.

Against these advantages there is, however, one serious objection to the use of separate liners. Whether the liner be pressed into the cylinder-jacket casting or bolted to the combustion head there is bound to be a great thickness of metal at the junction, and since

this is a point at which the temperatures are exceedingly high it is a most undesirable feature. It is not proposed to generalize any further upon mechanical details, because these must necessarily depend to a great extent upon the type and size of engine, but the principal mechanical features of each of the leading types of engine will be considered when dealing with them in detail.

CHAPTER XV

BALANCING

One of the most important considerations in the design of all reciprocating engines is the question of balance, and in the case of high-speed engines, or engines which are mounted on light structures, such as aeroplanes, motor-cars, and boats, good balance becomes a vital necessity. The kinds of vibration are numerous, but, broadly speaking, they may be classified under two heads: (1), vibration due to the reaction of the impulses which tend to rotate the whole engine around its crankshaft, and (2), vibration due to changes in the position of the common centre of gravity of the reciprocating parts.

Considering the first cause, it is evident that the forces tending to rotate the engine around the crankshaft are precisely equal to those tending to rotate the crankshaft itself, and are dependent upon the fluid and inertia pressures within the cylinder. It is equally evident that if the number of cylinders were so multiplied that the turning moment were even throughout the whole cycle, there would be no vibration but merely a constant static pressure equal to the torque of the engine. From this it is clear that reactionary vibrations are dependent upon the turning moment, but they are also dependent upon the ratio of the inertia of the engine to the magnitude of the torque variations. The heavier the engine and the smaller the torque variations the less will be the vibration from this source. Time, also, is an important element, for although the forces tending to rotate the engine about the crankshaft may be the same at all speeds, the time during which they act is reduced with increase of speed, and hence the actual displacement of the engine is reduced.

From the above considerations it is clear that reactionary vibration is a function of the speed, number of cylinders, and weight of the engine, and is at a maximum in the case of single-cylinder, slow-running, four-cycle engines, and at a minimum in the case of high-

speed multiple-cylinder engines, especially of the two-cycle type, in which the torque variations are much smaller. Since this cause of vibration is largely dependent upon the fluid pressures within the cylinder it follows that it is also dependent upon the load of the engine (unless hit-and-miss governing be employed), and is at a minimum when the engine is running light.

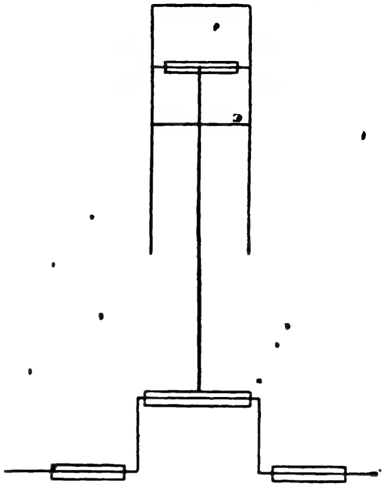


Fig. 52

The second cause of vibration, namely, that due to the displacement of the centre of gravity of the reciprocating parts, is the more serious of the two, because it increases with the square of the speed and is entirely independent of the load. Moreover, in order to obtain a light and efficient engine, a comparatively high rotational speed is desirable, and it is of the utmost importance that the reciprocating parts should be as well balanced as possible; in other words, that the common centre of gravity of all the reciprocating parts should be stationary or as nearly stationary as possible. Figs. 52, 53 show two arrangements of cylinders and cranks that are commonly employed. Fig. 52 is a single-cylinder engine either vertical or horizontal. With such an engine, running on the four-stroke cycle, the reactionary vibration is very serious, since there is only one impulse every fourth stroke. Consequently the interval between the impulses is very great and their magnitude in proportion to the total

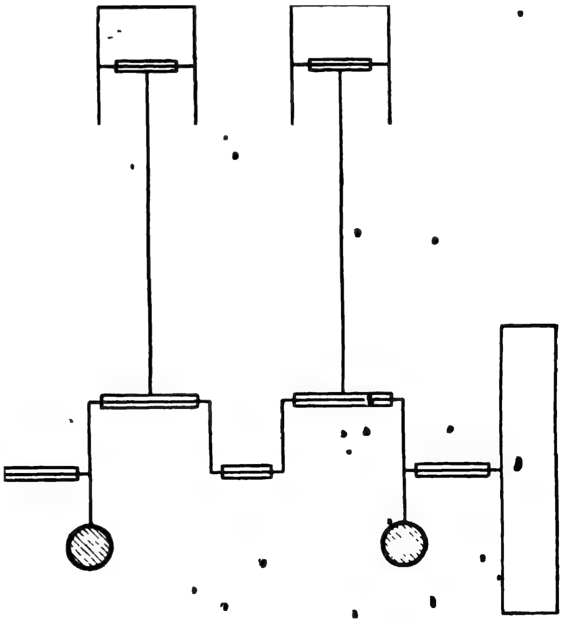


Fig. 53

weight of the engine is also great. When running on the two-stroke cycle the reactionary vibration is not nearly so great, because the interval between the impulses is shorter and the weight of the engine generally somewhat greater in proportion to their magnitude, but in either cycle the reactionary vibrations are at a maximum in single-cylinder engines. Vibration due to the displacement of the centre of gravity of the reciprocating weights is also very serious. For the sake of simplicity, suppose that the centre of gravity of the reciprocating parts be coincident with the gudgeon-pin of the piston, then the displacement of the centre of gravity in this case is equal to the displacement of the piston, and the forces tending to cause vibration are equal to the inertia forces of the piston. Such an engine cannot be balanced as regards the reciprocating parts, though the rotating parts can be completely balanced by the addition of balance weights to the crank.

Fig. 53. In this arrangement, since the two pistons are connected to the same crank, the impulses, in the case of a four-cycle engine, are equally distributed, and the turning moment is the most uniform obtainable from two cylinders. Consequently the reactionary balance is the best obtainable, but the reciprocating balance is bad, because, since the two pistons reciprocate together, the displacement of their common centre of gravity is the same as in a single-cylinder engine. For a given power of engine the actual reciprocating balance is, however, better than that of a single-cylinder engine, because, other things being equal, the power varies as the square of the diameter of the piston, while the weight of the reciprocating parts varies nearly as the cube of the diameter. Since reactionary vibrations are at their maximum at low speeds, and reciprocating vibrations at high speeds, it is clear that this arrangement of cylinders and cranks is more suitable for slow-running engines.

The arrangement shown in fig. 54 is practically the converse of fig. 53 as regards balancing. In this arrangement, in the case of four-cycle engines, two impulses follow one another successively; there is then an idle period during two strokes, consequently the turning moment is very poor, for the period of high torque extends over two strokes and the displacement of the whole mass of the engine is very great. From the point of view of reactionary balance, this is probably the worst possible arrangement. On the other hand, in so far as reciprocating balance is concerned, it is fairly good. The two pistons travel up and down in opposite phase,

consequently the displacement of their common centre of gravity is only that due to the angularity of the connecting-rods.

To appreciate this, suppose that both cranks were at 90° to the dead centre, then one piston is travelling downwards and the other upwards, but neither of them is at its mid-stroke position unless the connecting-rods are infinitely long—the shorter the connecting-rods the greater the distance of the pistons below mid-stroke. When either piston is at the top of the stroke the common centre of gravity is at mid-stroke, and midway between the two; but when

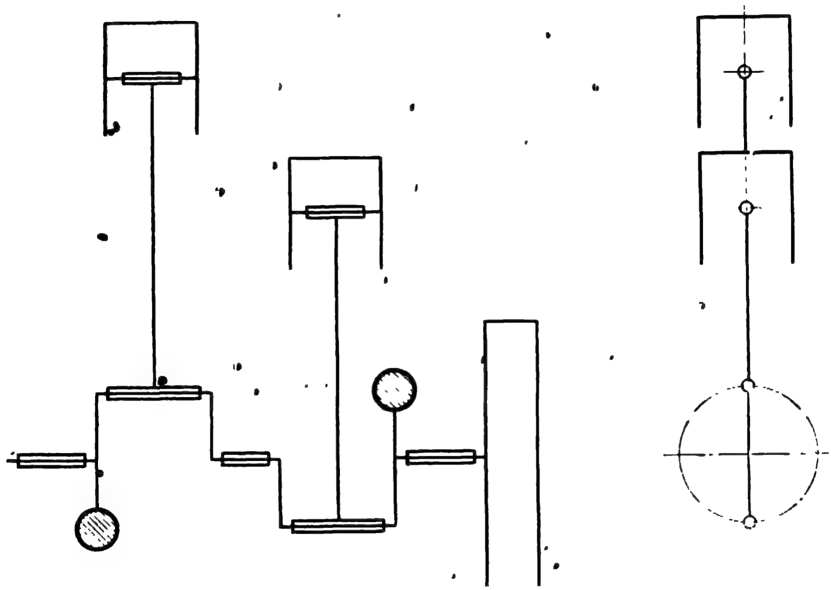
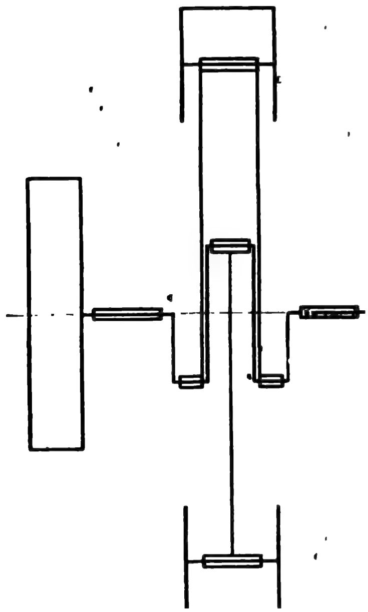


Fig. 54

the crank has been rotated through 90° both pistons are beyond the mid-position, and the common centre of gravity has been displaced by an amount depending upon the ratio of the length of the connecting-rod to the length of the crankthrow. If now the crank be rotated through a further 90°, both pistons arrive at their dead centres again, and the common centre of gravity returns to the mid-stroke position. This displacement of the centre of gravity, due to the angularity of the connecting-rod, is not usually very serious, and is dependent upon the length of the rod, but it occurs twice every revolution. Such vibrations are generally known as secondary or octave vibrations.

In addition to the octave vibrations there is also an unbalanced couple due to the distance apart of the two cylinders, and this

produces serious vibrations in a fore-and-aft direction along the crankshaft. This couple can be mitigated to a certain extent, but not entirely eliminated, by fitting suitable balance weights to the crankwebs. Compared with figs. 52 and 53 this arrangement is far better balanced as regards the reciprocating parts, for the secondary vibrations and the couple produce disturbances that are comparatively small. For four-cycle engines this is probably the best arrangement where high speeds are employed, and in which the very imperfect reactionary balance is not severely felt. For two-cycle engines it is excellent. In this case the turning moment is as uniform as can be obtained with two single-acting cylinders, for there is an impulse at each stroke.



Fig

In this instance the primary disturbing forces are balanced, but the secondary forces are not, and there is also an unbalanced couple.

The arrangement shown in fig. 55, in which two horizontal opposed four-cycle cylinders have their pistons connected to crank-pins at 180° to one another, is the best possible arrangement for a two-cylinder engine from the point of view of balance. From the point of view of reactionary balance the engine is equal to that shown in Fig. 53, and there is one impulse during each revolution; consequently the interval between the impulses is reduced to a minimum, and the turning-moment is as regular as is obtainable from any two-cylinder four-cycle engine. From the point of view of reciprocating balance the engine is perfect, for the position of the common centre of gravity is always coincident with the centre of the crankshaft, and undergoes no displacement whatever. Since both pistons are travelling in the same phase, and not alternately, it follows that the error due to the angularity of the connecting-rods is disposed of, and the engine completely balanced, both as regards primary and secondary vibrations.

The employment of a three-throw crankshaft, however, is somewhat expensive, and since the crankwebs must necessarily be very

stiff and heavy, it is difficult to provide an adequate amount of bearing surface on the crankpins. For this reason engines of this type are generally constructed as shown in fig. 56, in which only a two-throw crank is employed, and the cylinders placed slightly out of line with one another. This construction, of course, introduces a slight couple, due to the distance apart of the centre lines of the two cylinders; the magnitude of this couple is very small compared with fig. 54, and except in the case of exceptionally high-speed engines, can generally be ignored. There is a great deal to be said for the arrangement shown in fig. 56 for four-cycle engines, and engineers do not appear to appreciate fully all the advantages to be gained by it. Accurate balance is a feature of very great importance, and will become even more so, if, as the author confidently expects, the rotative speed of internal-combustion engines is increased in the near future. The principal objections are: (1) That it is only practicable in the case of horizontal engines; (2) the engine is bulky and takes up a great deal of floor space; (3) the combustion chambers are a long way apart, involving the necessity for lengthy and somewhat complicated pipework, and making the inspection and supervision of the valves and valve gear somewhat troublesome; (4) it is not applicable

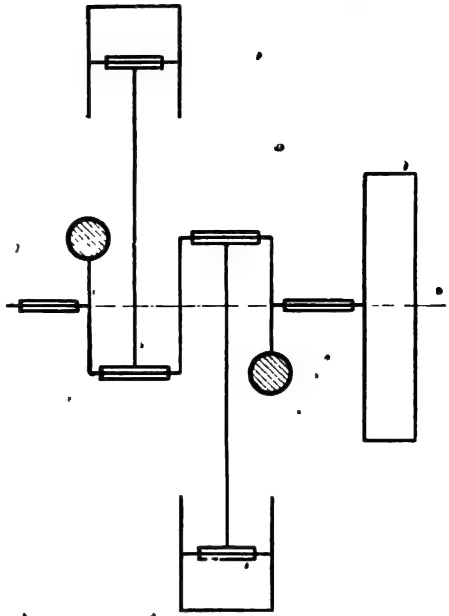


Fig. 56

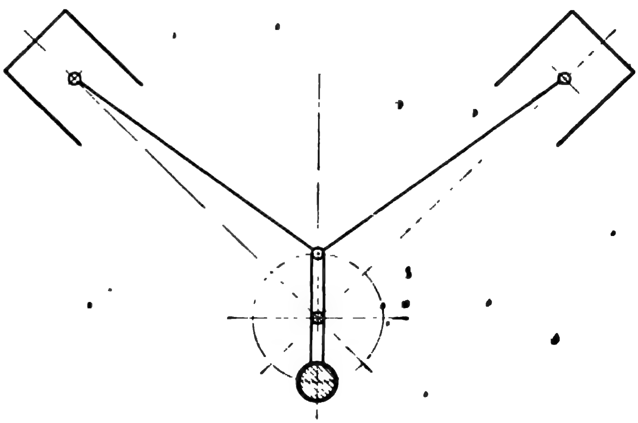


Fig. 57

to engines running on the two-stroke cycle, because in that case both pistons would necessarily be making their expansion strokes simultaneously, and the advantage of a more even turning moment, one of the most valuable assets of a two-cycle engine, would be lost.

In fig. 57 is shown an arrangement frequently adopted for motor-cycle engines, in which two cylinders are placed at an angle of 90° to one another, and both pistons are connected to a single crankpin. This arrangement gives an irregular turning moment midway between figs. 53 and 54, hence the reactionary vibration is also midway between these two extreme cases. If suitable balance

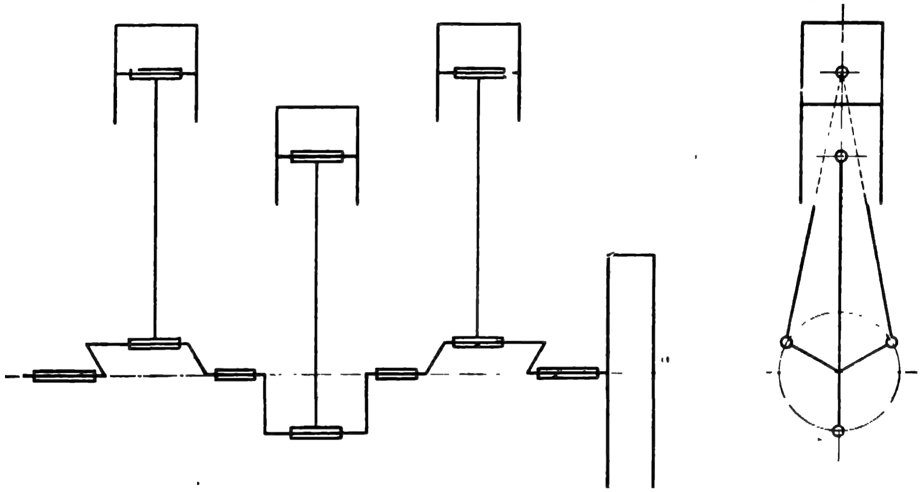


Fig. 58

weights are fitted, the primary reciprocating balance is perfect, and there would be no displacement of the centre of gravity if the two pistons travelled in true harmonic motion, that is to say, if the two connecting-rods were infinitely long. There is, however, a slight displacement, due to the angularity of the connecting-rods, and therefore there will be octave or secondary vibrations depending upon the length of connecting-rods employed. Since the two cylinders can both be arranged in the same plane at right angles to the crankshaft, there is no unbalanced couple along the shaft. Compared with other two-cylinder arrangements this is better than figs. 53 or 54, but inferior to figs. 55 or 56.

For three-cylinder engines, whether two- or four-cycle, single- or double-acting, the arrangement shown in fig. 58 is invariably adopted. In this arrangement all three cylinders are placed in a row along the crankshaft, and the three pistons are connected to

cranks equally spaced at 120° to one another. By this means all primary and secondary forces are perfectly balanced, and the turning moment is as regular as can be obtained from three cylinders. Unfortunately, however, there is a very large unbalanced couple along the crankshaft, which introduces serious vibration in a fore-and-aft direction, and can only be minimized by placing the cylinders as close together as possible and, to some small extent, by the employment of balance weights. But for this unbalanced couple, this form of three-cylinder engine would be almost vibrationless, for there is no primary or secondary vibration, and the reactionary balance is the best obtainable from any arrangement of three cylinders.

For four cylinders operating on the four-stroke cycle the arrangement shown in fig. 59 is almost invariably adopted, and since almost

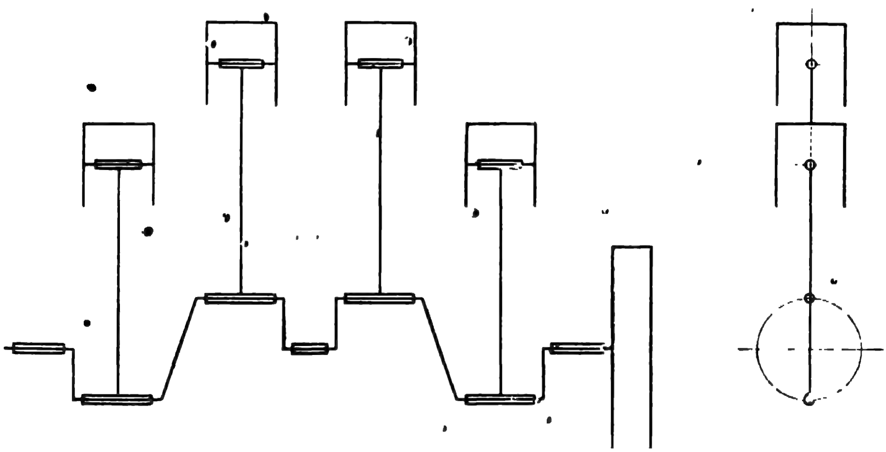


Fig. 59

all automobile, and a very large proportion of the marine and stationary engines, are now built with four cylinders arranged in this manner, it is worth while to consider it in some detail. The four cylinders are arranged in a row along the crankshaft, and the pistons are connected to four crankthrows at 180° to one another, that is to say, all the four cranks are in one plane. The two inside cranks are together, and form in effect one crankpin, and the two outside cranks are also, of course, together. From the point of view of reactionary balance the arrangement is the best possible, for the impulses are equally spaced, and there are two during every revolution.

The whole engine should be regarded as two engines of the type shown in fig. 54, coupled together in such a manner that the two

impulses of one engine occur during the two idle strokes of the other; the two unbalanced couples tending to rock each component half of the engine in opposite directions are opposed, so that they exert a bending strain upon the structure of the complete engine, but do not tend to produce any external movement. Mr. Lanchester very aptly christens engines such as these, in which the unbalanced couples of the two component parts neutralize one another, "looking-glass" engines.

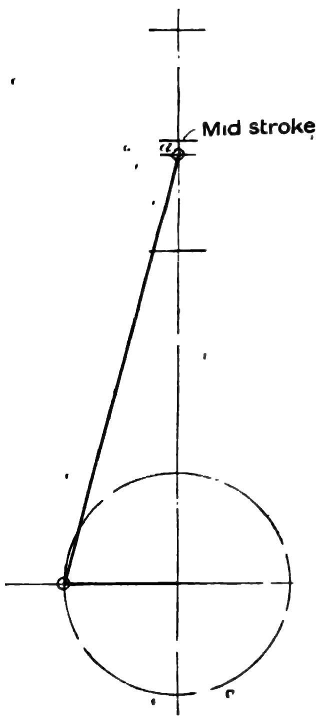


Fig. 60

Since the two inside and the two outside pistons reciprocate alternately, it follows that the primary forces are balanced, and there is no primary vibration. There is, however, an octave or secondary vibration, and this assumes very large proportions, for, since all four cranks are in one plane, it follows that at about mid-stroke all four pistons are simultaneously displaced to a point beyond the midstroke position. The amount of this displacement depends, of course, upon the relative length of the connecting-rods.

Referring to fig. 60, the midstroke displacement of the pistons due to the angularity of the connecting-rods is represented at *a*, and *a* is approximately equal to the crank-throw squared, divided by twice the length of the connecting-rod. Suppose, that in an engine of 2-ft. stroke the connecting-rod were only 4 ft. long, then the ratio of length of connecting-rod to crank-throw would be as 4 : 1, and the displacement *a* would be

$$1^2 \div 8 = \tfrac{1}{8} \text{ ft., or } \tfrac{1}{16}\text{th of the piston's stroke}$$

Thus the amplitude of the secondary vibration is equal to one-sixteenth of the stroke. Now, since the force is proportional to the square of the speed, and the secondary vibrations occur twice as rapidly as the primary, it follows that the maximum disturbing force when compared with the primary force is

$$2^2 \times \tfrac{1}{16} = \tfrac{1}{4}\text{th.}$$

That is to say, the disturbance created by the secondary vibrations is equal to one-fourth of that created by the primary vibrations in a single-cylinder engine, in which the length of the connecting-rod is equal to twice the stroke.

Now it has already been shown that in a four-cylinder engine of this type the secondary displacement of all four pistons occurs simultaneously, so that the total secondary disturbance must be multiplied by four and is equal to

$$\frac{1}{4} \times 4 = \text{unity.}$$

In other words, the secondary vibration in an engine of this type, and with this ratio of connecting-rod length, is exactly equal to the primary vibration in a single-cylinder engine, but the external vibration is not so great, because the weight of a four-cylinder engine is necessarily nearly four times as great as that of a single-cylinder engine of the same cylinder dimensions. The length of connecting-rod taken in this instance is, of course, considerably shorter than would be used in any well-designed engine, but it serves to illustrate how serious secondary vibrations may become, and how important it is to employ as long a connecting-rod as possible. Because all couples and all primary forces are balanced, and the turning moment is extremely good, it is popularly supposed that the balance of engines of this type is particularly good. How great is the magnitude of the unbalanced secondary disturbances, and how important an influence the length of the connecting-rod has in this particular case are not generally appreciated.

Balance of Four-cylinder Two-stroke Engines.—For four-cylinder engines of the two-cycle type, either of the arrangements shown in figs. 61 and 62 may be adopted. The former arrangement is generally the most convenient. In this, all four cylinders are placed in a row along the crankshaft, and the pistons connected to cranks at 90°. Each end pair of cranks are at 180° to one another, but the two middle cranks are at 90°. The engine is in effect two units of the fig. 54 type coupled together, with their crankshafts at 90° instead of 180°. It is thus not a "looking-glass" engine, and the couples of the two component parts are out of phase, and do not completely neutralize one another. The turning moment is, of course, extremely regular, for there are four equally spaced impulses per revolution, so that reactionary balance is reduced to a minimum. The primary disturbing forces are, of course, balanced, and the secondary also, but

in this case there is a couple due to the midstroke displacement of the pistons occurring alternately in each pair of cylinders. Compared with fig. 58, however, the vibration from this source is trifling. With this arrangement, balance weights should be used to com-

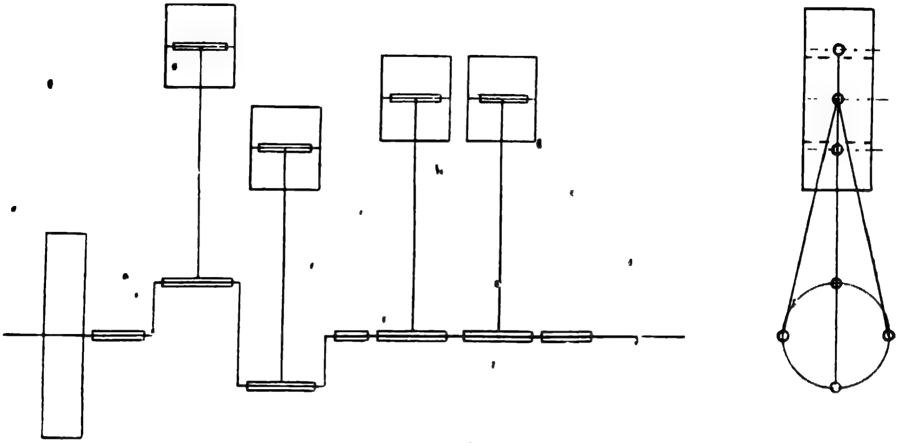


Fig. 61

pletely balance the rotating parts, and mitigate as far as possible the unbalanced couples.

The arrangement shown in fig. 62 is in many ways preferable to that shown in fig. 61, but there are a great number of purposes for which inclined cylinders are unsuitable on account of the width and

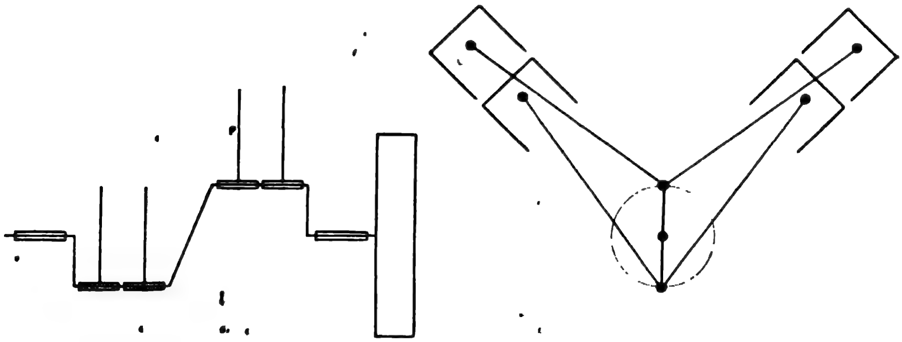


Fig. 62

inaccessibility of the engine. In this arrangement, two pairs of cylinders are arranged at 90° to one another, the opposite pistons of each pair being coupled to the same crankpin, and the two cranks placed at 180° to each other. In so far as reactionary balance is concerned, there is nothing to choose between the two types, both are as good as it is possible to obtain from four cylinders. This

however applies only to the case of the two-cycle engine. The primary disturbing forces are completely balanced, and it is also possible completely to balance the couple along the crank, except for a very slight error due to secondary disturbances, for in this particular case the unbalanced components of the couple can be not only minimized, but almost completely neutralized, by the addition of balance weights to the crankshaft. The reason for this is that in a **V**-type engine at 90° the locus of the common centre of gravity of any two opposite pistons is an almost circular path, consequently the couple is rotating, not reciprocating, and can therefore be neutralized by means of rotating balance weights.

Fig. 62. The general reciprocating balance obtained by this

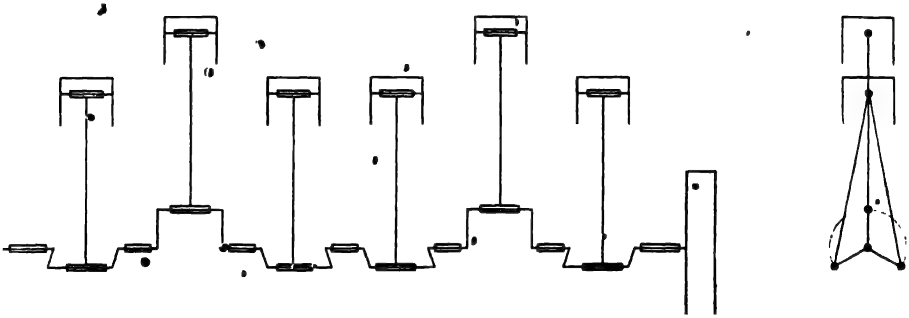


Fig. 63

arrangement is very good, but it can, of course, only be used to advantage for two-cycle engines, since the impulses would not be equally spaced if the cylinders operated on the four-stroke cycle.

Balance of Six-cylinder Engines.—For six-cylinder engines, the arrangement shown in fig. 63 is almost invariably employed. In this, the six cylinders are arranged in a row along the crankshaft, and the pistons connected to cranks at 120° to one another. It consists, in effect, of two three-cylinder engines coupled together, and, as in the four-cylinder arrangement, fig. 59, the two central pistons are connected to a common crankpin. The two intermediate and the two outside pistons also reciprocate together. It is another example of a "looking-glass" engine, in which the reciprocating couples of the two three-cylinder units are opposed to one another, so that they tend to bend the engine about the central point, but not to cause any displacement of the whole mass. The turning moment and the reactionary balance are the best possible for this number of cylinders. Both the primary and secondary

disturbing forces are balanced, as in the three-cylinder engine, and at the same time the one great defect of that type, namely, the large unbalanced couple, is eliminated. It would appear, therefore, that the balance obtained by this arrangement should be almost perfect, and that the engine should be capable of running at any speed without vibration. There are, however, other causes affecting the vibration of an engine which will be dealt with shortly, and which have a particularly deleterious effect upon six-cylinder engines.

Eight-cylinder "V" Motors.—Eight-cylinder four-cycle V-type engines have recently been extensively employed in connection with aeroplanes, and the arrangement usually adopted is shown in fig. 64. In this, two sets of four cylinders are arranged at 90°

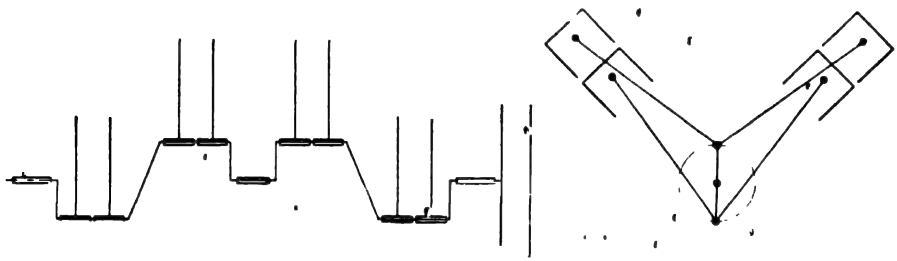


Fig. 64

to one another, and the pistons connected to a four-throw crankshaft, with all cranks in one plane, the two opposite pairs of pistons being connected to the same crankpin. By this means a regular turning moment is obtained, and the impulses are equally spaced, there being four per revolution. The primary balance is perfect, but, as in the four-cylinder vertical engines, the secondary disturbing forces are cumulative, and may be represented by the horizontal diagonal of a square in which the components of the disturbing forces of the two sets of cylinders are represented by two of the inclined sides. The total secondary disturbing force is therefore equal to

$$4 \times \sqrt{2} = 5{\cdot}65;$$

that is, 5·65 times the secondary disturbing forces of a single-cylinder engine. In other words, the secondary vibrations in engines of this type are 41 per cent worse than in a four-cylinder vertical engine, and the need for long connecting-rods to reduce the magnitude of these secondary disturbing forces is clearly very great.

CHAPTER XVI

DOUBLE-PISTON ENGINES—SYNCHRONOUS VIBRATION

Double-piston Engines.—In addition to the arrangements already considered, there is also another class in which two pistons are employed in one cylinder, the principal object being: (1) to obviate the use of a combustion chamber, always a source of weakness, especially in very large engines; (2) to obtain a better balance than is possible with one piston only; (3) to reduce the weight of the engine by relieving the framework from stress.

In the arrangement shown in fig. 65 the cylinder is in the form of a plain, open-ended tube, and is provided with two pistons which reciprocate in opposite directions, and are connected to cranks at 180° to one another, the outer piston being connected by means of long return rods. In this arrangement the reactionary balance is, of course, exactly the same as in a single-cylinder single-piston engine, for this depends upon the number and spacing of the impulses per revolution, and is unaffected by the number or relative motion of the pistons. In so far as reciprocating balance is concerned, since the pistons are travelling in opposite phase, it follows, as in the arrangement shown in fig. 55, that all primary disturbing forces are balanced, but the secondary forces are in this case unbalanced and cumulative. The reciprocating weight, however, is very great, and this, as has already been shown, has a most detrimental influence on the mechanical efficiency.

The arrangement shown in fig. 66 is employed in the Fullagar engine, a two-cycle engine which the author believes has considerable promise of success, in that it contains many of the advantages and eliminates some of the disadvantages of the last-mentioned type. In this engine two open-ended cylinders are arranged side by side, and each contains two pistons coupled in pairs by means of diagonal tie-rods as shown. The two lower pistons are connected directly to a two-throw crankshaft with cranks at 180°. By this

means the two sets of pistons converge alternately, and there is one impulse at each stroke, or two per revolution, so that the turning moment and reactionary balance are the best obtainable from the number of cylinders. All primary forces are completely balanced and there is no unbalanced couple, for the couple due to the upper

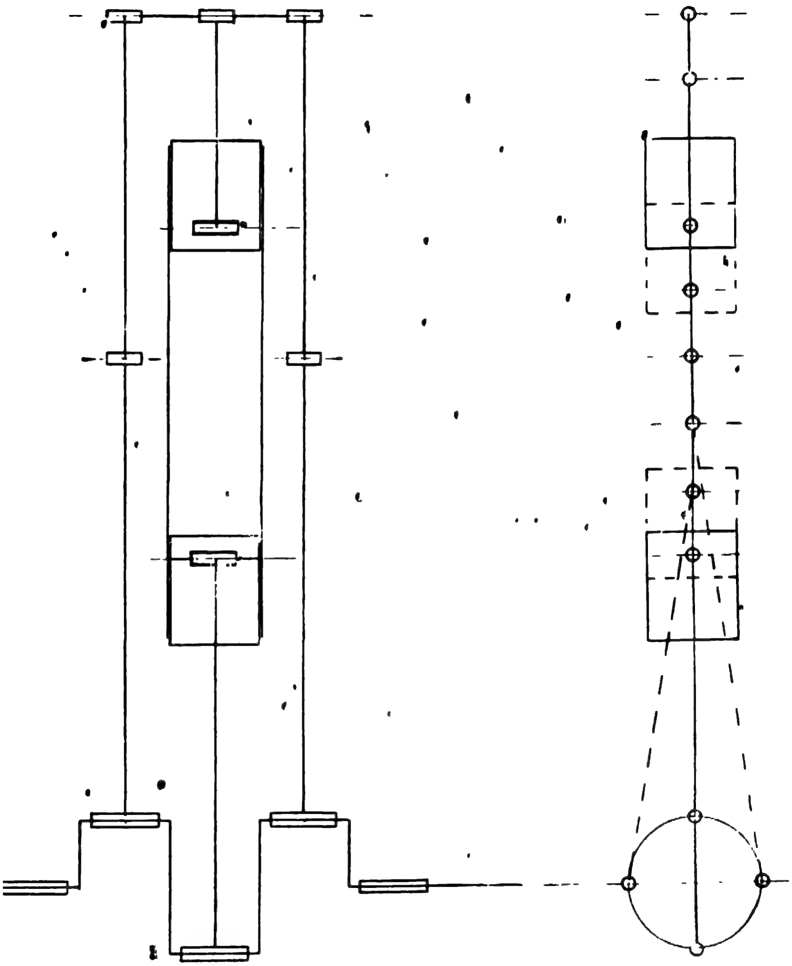


Fig 65

pair of pistons is neutralized by that due to the lower. The secondary forces, however, are unbalanced, and are cumulative as in the four-cylinder arrangement fig. 59, for, when the two cranks are in a horizontal plane, all four pistons are simultaneously displaced below the midstroke position. The balance of this engine is in every way identical with that of a four-cylinder four-cycle engine. The unbalanced secondary forces can be counteracted by the addi-

tion of a second pair of cylinders with cranks at 90° to the first, but in this case there still remains a small secondary couple. To eliminate octave disturbances entirely it would be necessary to add two more pairs of cylinders and make a "looking-glass" engine of it.'

The arrangement shown in fig. 67 is sometimes employed. Without going into details it is evident that the reactionary balance is the same as in an ordinary single-piston engine, and that the reciprocating balance is perfect both as regards primary and secondary disturbing forces.

The arrangement shown in fig. 68 is becoming very popular for two-cycle engines on the grounds of the excellent facilities it affords for scavenging. Its balance, however, is its weakest feature. In so far as reactionary balance is concerned it is the same as a single-cylinder single-piston

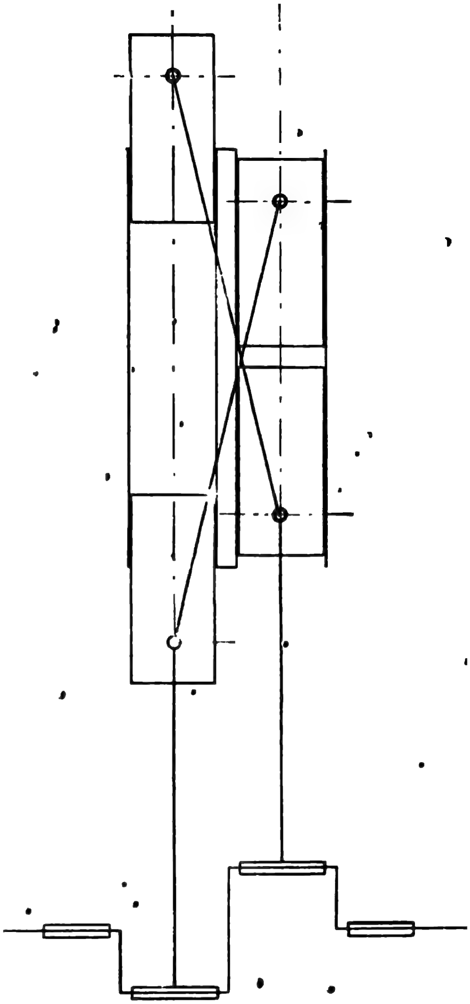


Fig. 66

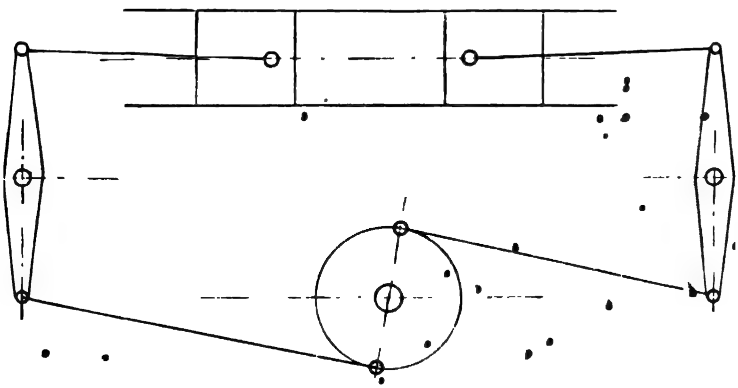


Fig. 67

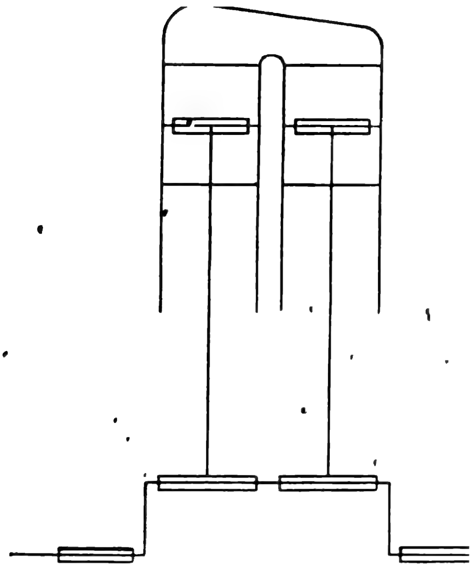


Fig. 68

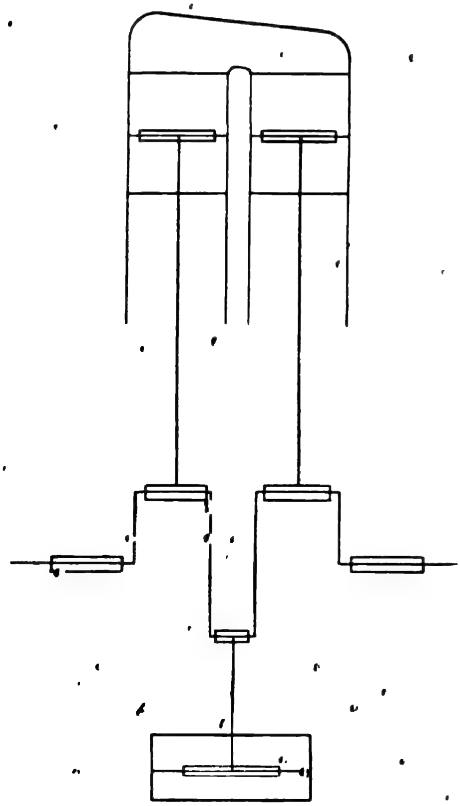


Fig. 69

engine, but, as in the arrangement shown in fig. 53, all primary forces are entirely unbalanced, and to obtain anything approaching a reciprocating balance the two cylinder units must be treated as single cylinders and grouped in some such manner as already indicated.

The arrangement shown in fig. 69 has been adopted by the author for small two-cycle engines, in which the two upper pistons form the power pistons and the lower one is used as a supplementary scavenge pump, as described in the volume dealing with two-cycle engines. In this arrangement the reactionary balance is the same as in a single-cylinder single-piston engine, but a perfect reciprocating balance, both primary and secondary, is obtained as in the two-cylinder arrangement fig. 55, provided, of course, that the weights of the reciprocating parts and the ratio of the connecting-rod length to crank-throw are the same in all cases.

Engines employing Reverse Rotation. — Some designers have from time to time made attempts completely to balance the reactionary forces by the employment of two crankshafts rotating in opposite directions, so that the reactions around the two shafts are equal

and opposite and exert stresses upon the structure but no displacement of the whole mass. Since, however, with modern high-speed engines, reactionary vibrations are of secondary importance compared to those set up by the reciprocating masses, the advantages to be gained by eliminating them do not seem great enough to justify the complication, weight, and expense of employing two crankshafts with two fly-wheels and the mechanical difficulties of gearing them together. Two interesting and, for a time, very successful engines have, however, been constructed on this principle.

The arrangement shown in fig. 70 is employed in the Lucas engine, which operates on the two-stroke cycle. In this engine two pistons are employed in a single inverted U-type cylinder, and each is connected to a separate crankshaft. The two shafts are geared together in such a manner that the pistons reciprocate synchronously. By this means it is clear that all reactionary disturbances are avoided. Also, by this method the primary disturbing forces can be balanced by means of balance weights attached to each crankshaft, for, referring to fig. 71, it will be seen that the common centre of gravity of the two balance weights travels up and down in a vertical plane midway between the two cylinders, and so balances the common centre of gravity of the two pistons; the secondary disturbing forces are not, however, eliminated, and, as in the arrangement shown in fig. 59, are cumulative. This engine, therefore, is perfectly balanced in all respects except as regards the secondary disturbing forces, which, being cumulative, are rather serious.

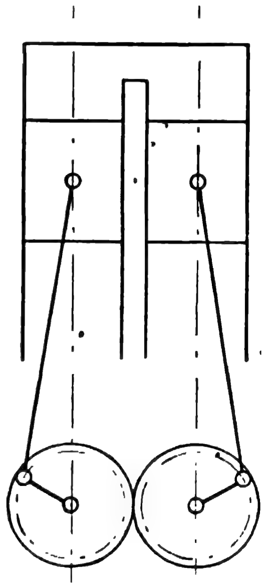


Fig. 70

The arrangement shown at fig. 72 is the Lanchester engine, which claims the distinction of being probably the only completely balanced reciprocating engine ever built. In this engine two horizontal opposed cylinders are employed, and the two pistons are each connected to two crankshafts as shown. These two crankshafts revolve in opposite directions, and the two pistons in opposite phase, consequently all reactionary, primary, and secondary disturbing forces are eliminated, and a mathematically perfect balance obtained. This engine was employed on the well-known Lanchester motor-cars for several years, but was ultimately abandoned on the grounds

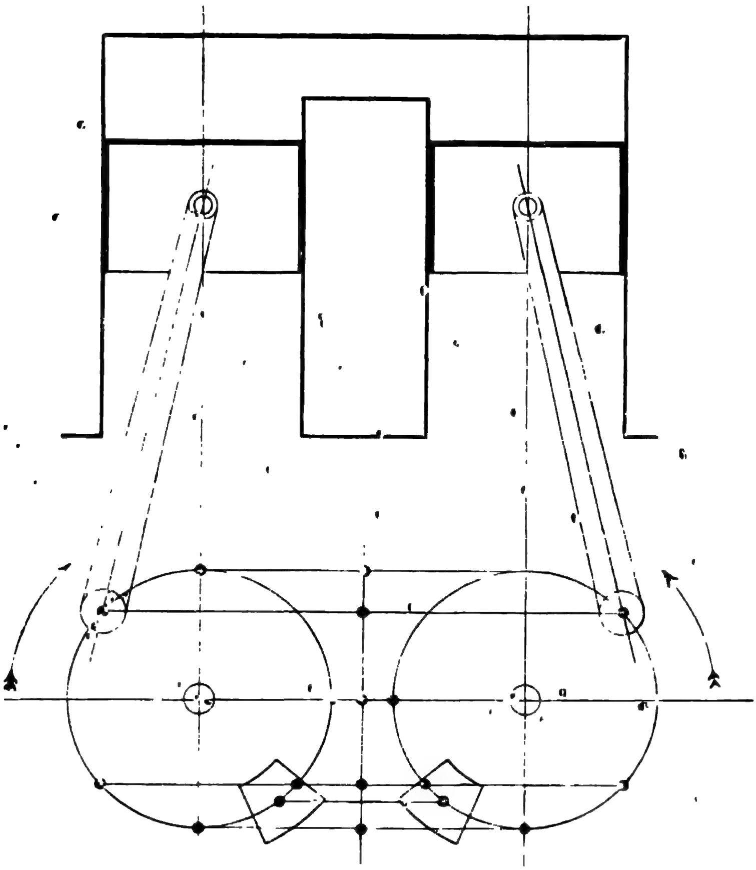


Fig. 71

of cost and complication, in favour of the vertical four- and six-cylinder arrangements.

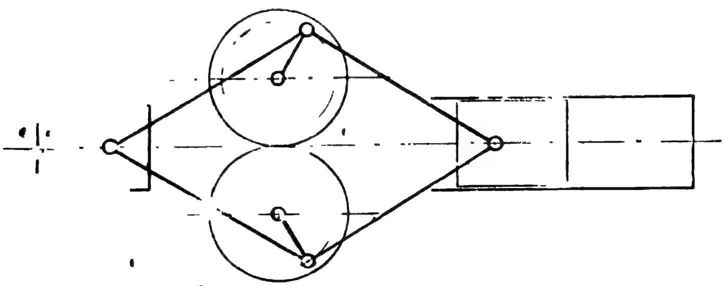


Fig. 72

Synchronous Vibrations.—There is yet a further source of vibration which has already been alluded to in the case of six-

cylinder engines. In this case the complete engine consists of two three-cylinder units, coupled together in such a manner that the two couples oppose one another and tend to bend the structure of the engine about the middle point, but do not tend to cause any displacement of the structure as a whole. Now, it is a comparatively simple matter to design the bed-plate and base chamber of a six-cylinder engine of such strength that there is little to fear from actual structural failure; the real difficulty arises when the natural vibration due to the elasticity of the structure coincides with the running speed, or with the period of any disturbances associated with the functioning of the engine. Thus, any natural vibration period in the engine structure may pick up the main piston period or the secondary period, or in some cases, the impulse period may be the exciting cause, so that synchronization may take place at several different running speeds of the engine.

At the times when synchronization occurs the amplitude of the vibrations is, of course, very greatly increased, and the whole engine will vibrate excessively. These synchronous speeds at which excessive vibration occurs are generally known as "threshing points" or "periods", and extend usually over a very narrow range of speed. That is to say, if the speed of the engine be increased or decreased by a comparatively small amount the threshing point will be avoided, and the engine will run without vibration. Periodic vibrations or threshing points can only be obviated by so stiffening the bed-plate and base chamber of the engine that the first synchronous speed does not occur until the engine is running at a higher speed than it is ever intended to do under normal conditions. To do this, in the case of high-speed six-cylinder engines, necessitates the use of very deep and heavy base chambers. In the author's opinion, it would be advisable so to design the cylinders of all six-cylinder engines that they can be bolted together to form one solid block, and thus greatly add to the stiffness of the entire structure. Such an arrangement would add to the difficulty of removing any but the outside cylinders, but this difficulty is not insuperable. In many motor-car engines of the six-cylinder type it is customary to cast the cylinders in two groups of three, an arrangement which has nothing to recommend it from the point of view of rigidity. Periodic vibrations of this kind are liable to occur in all engines of the "looking-glass" type, in which the unbalanced couples of the two or more component parts are opposed to one another, but they are particularly noticeable in the case of six-cylinder engines,

in which the magnitude of the couples is very much greater than in other arrangements.

There is also a further source of periodic vibration, due to the torsional elasticity of the crankshaft, and this is particularly noticeable in long-stroke engines. In any multi-cylinder engine, the fly-wheel may be assumed to rotate at a uniform angular velocity, but the crankshaft, at the end farthest from the fly-wheel, is to some extent "wound up" during the impulse stroke, and suddenly released at the end of it. Thus it will tend to fly back to beyond its original position, and continue in a state of vibration until the next impulse. At certain definite speeds the impulses will synchronize with the periodicity of the crankshaft, or some function of it, and the amplitude of the vibrations will be greatly increased in consequence. Vibrations set up in the crankshaft are not entirely self-contained, but are transmitted to the structure of the engine through the pistons and connecting-rods. This source of vibration cannot be overcome by increasing the stiffness of the crankshaft without a very serious addition of weight and friction.

Lanchester Vibration Damper.—To obviate this source of vibration, Mr. F. W. Lanchester has recently patented a most ingenious device, known as a vibration damper, whose function it is to damp out the torsional vibrations in the crankshaft. This device, illustrated in fig. 73, comprises a small fly-wheel, mounted on the crankshaft at the end opposite to the main fly-wheel. It is, however, not rigidly attached to the shaft, but is mounted on bearings, so that it can rotate independently of it. Driving connection between the shaft and the fly-wheel is maintained by means of a friction clutch of the multiple-disk type. The whole damper revolves with the crankshaft, but the fly-wheel portion of it has sufficient inertia to maintain a uniform angular velocity, and any sudden change in the angular velocity of the shaft, due to torsional elasticity, involves slipping of the friction clutch against a considerable resistance. This has the effect of rendering the crankshaft dead-beat. It does not prevent the crankshaft from "winding up", but it does prevent it from vibrating when released. So far as the author is aware, the Lanchester vibration damper is only used on motor-car engines, but there seems no reason why it should not be employed with advantage on all engines in which a long crankshaft is used. The weight of the damper fly-wheel is trifling, as also its diameter, for comparatively little inertia is required to check the torsional vibrations of the crankshaft.

These periodic vibrations, due to lack of rigidity in the structure of the engine and crankshaft, are of little consequence in engines of less than six cylinders, but in this type they become very serious indeed, so much so, in fact, that many six-cylinder engines set up at certain speeds vibrations which are far worse than any produced by a four-cylinder engine. It is impossible to tabulate all the different arrangements described above in order of balance, for so

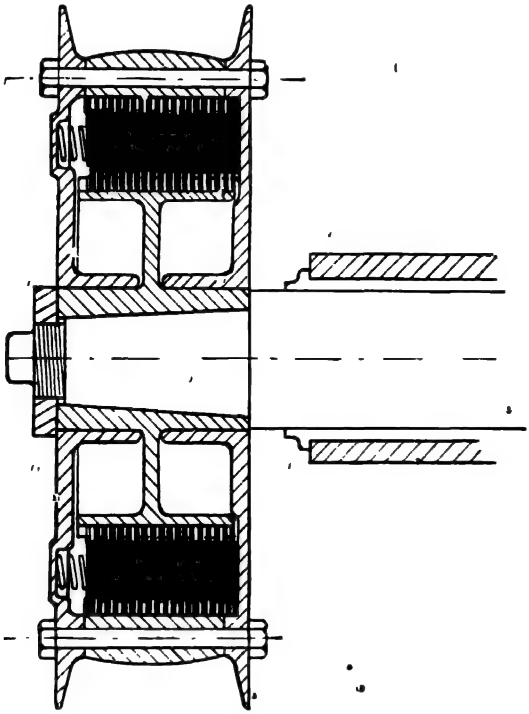


Fig. 73

much depends upon such factors as connecting-rod length and rigidity, but the following table summarizes the conclusions arrived at.

By way of comparison, it may be stated broadly that unbalanced secondary forces produce disturbances whose intensity is generally about one-sixth as great as unbalanced primary forces. No comparison is given as to rigidity, but this must depend both upon the length of the crankshaft and upon the couples which are opposed. Broadly speaking, the shorter the crankshaft the less the danger of periodic vibrations or threshing points.

BALANCING—SUMMARY OF TYPES

No. of Fig. Denoting Arrangement.	Reactionary Disturbing Forces, Ratio of Total Number of Strokes to Impulse Strokes.	Primary Disturbing Forces	Secondary Disturbing Forces.	Disturbing Forces due to Couple.
52 (four-cycle)	1·0	unbalanced	unbalanced	none.
52 (two-cycle)	2·0	unbalanced	unbalanced	none.
53 (four-cycle)	2·0	unbalanced (cumulative)	unbalanced (cumulative)	none.
54 (four-cycle)	irregular	balanced	unbalanced (cumulative)	unbalanced.
54 (two-cycle)	1·0	balanced	unbalanced (cumulative)	unbalanced.
55 (four-cycle)	2·0	balanced	balanced	none (perfect balance of mass).
56 (four-cycle)	2·0	balanced	balanced	very small.
57 (four-cycle)	irregular	balanced	unbalanced	none.
58 (four-cycle)	1·5	balanced	balanced	unbalanced (very large).
58 (two-cycle)	0·66	balanced	balanced	unbalanced (very large).
59 (four-cycle)	1·0	balanced	unbalanced (cumulative)	none
61 (two-cycle)	0·5	balanced	balanced	small secondary couple.
62 (two-cycle)	0·5	balanced	unbalanced	none.
63 (four-cycle)	0·66	balanced	balanced	none (perfect balance of mass).
64 (four-cycle)	0·5	balanced	unbalanced (cumulative)	none.
65 (four-cycle)	4·0 (per cylinder)	balanced	unbalanced (cumulative)	none.
65 (two-cycle)	2·0 (per cylinder)	balanced	unbalanced (cumulative)	none.
66 (two-cycle)	1·0 (per unit of two cylinders)	balanced	unbalanced (cumulative)	none.
67 (four-cycle)	4·0 (per cylinder)	balanced	balanced	none (perfect balance of mass).
68 (two-cycle)	2·0 (per unit)	unbalanced	unbalanced	none.
69 (two-cycle)	1·0 (per cylinder)	balanced	balanced	none (perfect balance of mass).
70 (two-cycle)	none	balanced	unbalanced (cumulative)	none.
72 (four-cycle)	none	balanced	balanced	none (perfect balance of mass and torque).

CHAPTER XVII

TWO-CYCLE ENGINES—GENERAL CONSIDERATIONS

The fundamental principles of the two- and four-stroke cycles have been dealt with in the first chapter of this volume. Both systems have very clearly marked advantages and disadvantages, but it would be rash to attempt to prophesy which will ultimately triumph; much, of course, depends upon the purpose for which the engine is required, and upon the heat-cycle which is employed.

Before comparing the two systems it will be best in the first place to discuss the various forms of two-cycle engine, of which there are a great variety.

In all two-cycle engines the last 20 per cent of the expansion stroke and the first 20 per cent of the compression stroke are devoted to the process of expelling the products of combustion and introducing the fresh charge, either of pure air or of combustible mixture. In other words, the two idle, or pumping, strokes in the four-cycle engine are done away with, at the expense of a small proportion of the expansion and compression strokes. To effect this, ports are arranged in the cylinder wall, which are uncovered by the piston towards the end of its stroke, and through which the products of combustion are allowed to escape, until the pressure within the cylinder has dropped to nearly atmospheric. Then, either a second series of ports, or a valve, is opened, and a charge of air, or in some cases combustible mixture, is introduced, generally by means of separate pumps. This charge drives out the remainder of the products of combustion and replaces them in the cylinder. It is in the method of supplying and introducing this scavenging charge that so great a variety exists. In the case of large engines, separate pumps are almost invariably employed, driven from the crankshaft of the engine, and either timed to deliver their charge while the exhaust ports are uncovered, or arranged to pump into a receiver, which in turn discharges into the cylinder at

the correct moment. In the case of smaller engines it is a common but objectionable practice to enclose the crankcase completely, and make use of the pressure created therein by the outward stroke of the piston for scavenging. This system, though largely employed, has very little to recommend it, for reasons that will be dealt with later.

From the pumps the charge enters the cylinder, either through valves arranged as far away as possible from the exhaust ports, or through a second series of ports on the side of the cylinder opposite

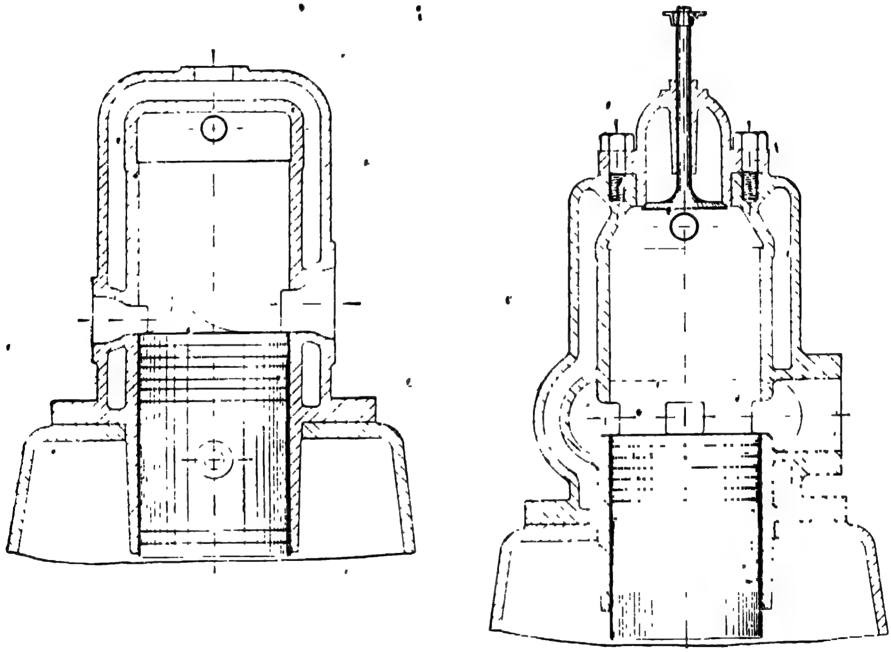


Fig 74a   Fig. 74b

to the exhaust. In the latter case, in order to prevent the charge from passing straight across the cylinder and out through the exhaust ports, it is usual to provide a deflector on the head of the piston, the function of which is to deflect the fresh charge upwards towards the head of the cylinder.

It will readily be seen that the success or otherwise of the two-cycle engine depends in a very large measure upon the efficiency of the scavenging, and the greatest care must be exercised in the design of all the scavenging arrangements to ensure that the maximum quantity of exhaust gases is expelled with the minimum loss of air. All the air employed has to be pumped to a pressure in excess of the residual pressure of the exhaust gases, an operation

which may involve the expenditure of a considerable amount of power, and obviously any escape of air through the exhaust ports involves the pumping of a larger quantity to make up the deficit. When the scavenging is effected by means of combustible mixture, as is sometimes the case, any loss through the exhaust ports becomes

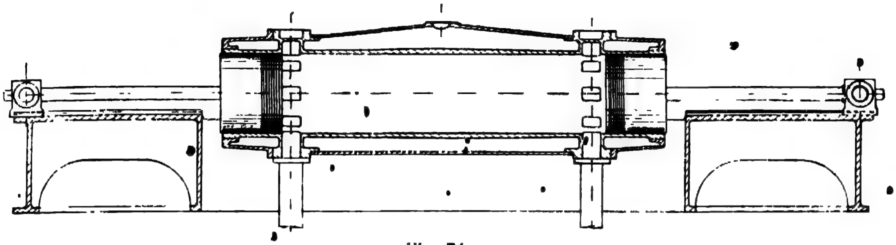


Fig. 74c

far more serious, for it is evident that any combustible mixture that escapes is a dead loss. In the best modern two-cycle engines, from 60 to 80 per cent of the exhaust gases is generally expelled with an expenditure of from 5 per cent to 9 per cent of the power of the engine. In some cases better results have been obtained, but generally at the expense of considerable complication and high initial cost. Figs. 74 *a*, *b*, *c*, *d* show diagrammatically some of the cylinder designs commonly adopted for scavenging.

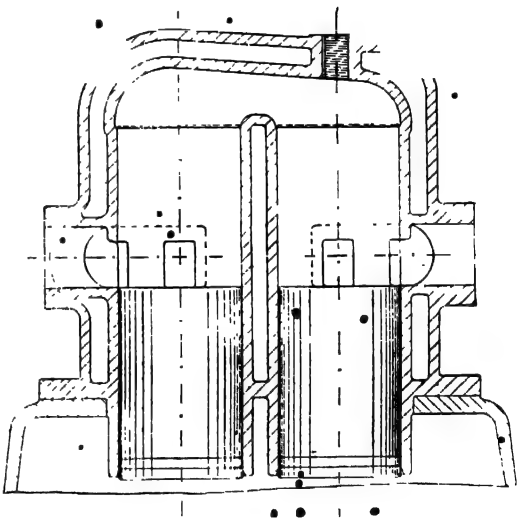


Fig. 74d

Scavenging Systems.—It will be well, before proceeding further, to consider these different systems in detail. In fig. 74*a* the same piston uncovers both the inlet and exhaust ports, and "short-circuiting is prevented, or at all events impeded, by means of the deflector on the head of the piston. As the piston descends on the expansion stroke it uncovers, first, the exhaust ports and then the inlet ports. Now it is evident that the exhaust ports must be opened considerably before the inlet ports, in order that a sufficient quantity of exhaust gases may have time to escape, and the pressure in the cylinder be reduced to something

below that supplied to the inlet ports; otherwise, instead of fresh air entering the cylinder, exhaust gases will pass back through the inlet ports, and foul the fresh charge before its entry. This means that either the exhaust ports must open so early as to lose a considerable portion of the expansion stroke, with consequent loss of efficiency, or the inlet ports must be very much restricted, so that a higher pressure is required to introduce the fresh charge. This latter, however, seriously increases the power absorbed in pumping, and limits the speed. Moreover, since both series of ports are on practically the same level, they must each be less than half the circumference of the cylinder in width; in practice, very much less, for the walls of the cylinder cannot be cut away indiscriminately. Again, on the upward stroke of the piston the inlet ports are closed considerably before the exhaust, and consequently a portion of the fresh charge is expelled through the latter, though this probably does not amount to very much. The system in its simplest form, though widely employed, is not a satisfactory one, the main objections being (1) insufficient port area, with consequent increased pumping losses; (2) exhaust opens too early, and a considerable proportion of the energy in the expanding gases is wasted; (3) too large a proportion of the cylinder wall is cut away in one plane, thus weakening the cylinder and causing unequal wear of the cylinder bore, due to the greatly reduced bearing surface of the piston-rings at this point.

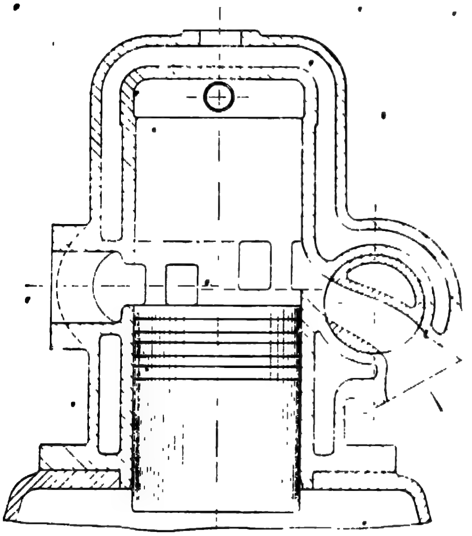


Fig. 75

A modification of this system which the author has advocated for some years, and which is now employed by several firms, is shown in fig. 75. In this case the inlet ports are uncovered by the piston slightly before the exhaust; but, to prevent the passage of exhaust gases back through them, they are masked either by a rotary or some other form of valve, until the pressure has dropped below the scavenging pressure. By this means the inlet ports can be made of slightly greater depth, and consequently greater area,

than the exhaust. Also, since on the compression stroke the inlet ports are still open after the exhaust are closed, an excess of air can be added if required, which is a very considerable advantage, for it is obvious that a very slight increase in the weight of charge at this point will raise the mean pressure considerably.

The addition of such a "delaying" valve, while removing many of the objections from this system, adds only a slight complication. Fig. 76 (A and B) show diagrammatically the timing of the port openings with and without the delaying valve, from which it will be seen that the effective inlet port area is doubled without en-

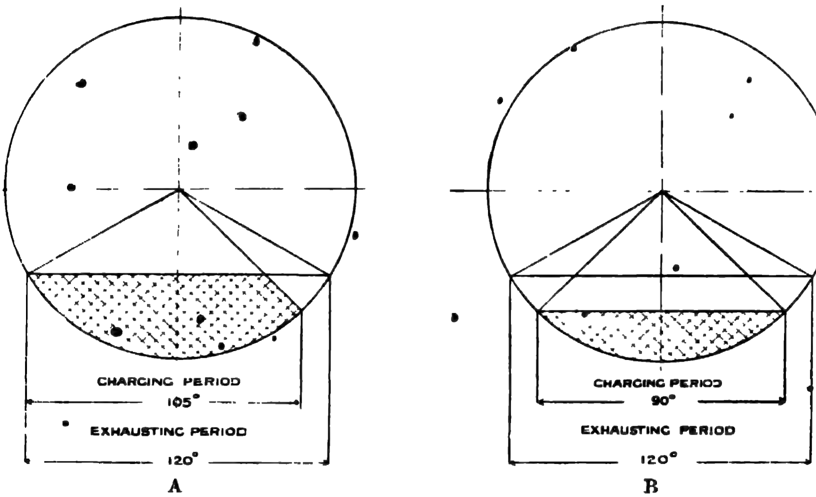


Fig 76

croaching any further on either the circumference of the cylinder or the expansion stroke.

Port or bottom scavenging, generally with a delaying valve, is now very extensively used for large Diesel engines, for it has the great advantage that it leaves the cylinder-head free from any valves except the fuel and starting valves, and so allows of much more efficient cooling of this most troublesome part. The system, however, is not suitable for gas-engines for two reasons: (1) The loss of fresh charge through the exhaust ports is necessarily considerable, and if gas were used it would be necessary to force it into the cylinder after the exhaust ports were closed, and consequently against a serious resistance. Where very rich gas was employed there would be no particular objection to this; but where, as is generally the case, the gas is of low calorific value, and consequently a large bulk is required, the power necessary to force it into the

cylinder would represent a serious loss. (2) It would be very difficult to govern the engine over widely varying loads, for it must be borne in mind that in a two-cycle engine the contents of the cylinder cannot be reduced by throttling as in a four-cycle engine, because at the commencement of the compression stroke the cylinder must always be full of air or of exhaust gases, or of a mixture of the two.

Governing by stratification, or qualitative governing, must therefore be relied upon for reduced loads; that is to say, the cylinder and the method of charging must always be so arranged that a small proportion of readily combustible mixture is undiluted and trapped in the neighbourhood of the igniter. With bottom scavenging, owing to the shape of the cylinder, and the diffusion which such a system of scavenging inevitably sets up, stratification is possible only to a very limited extent, unless a small separate pocket be provided, into which a small charge of combustible mixture can be pumped.

Scavenging by Valve in Cylinder Head.—The second method of scavenging, shown in fig. 74*b*, is the one now most generally adopted. In this system a ring of exhaust ports is provided round the circumference of the cylinder, and the fresh charge is admitted through one or more valves in the cylinder head. Engines using this system should have as long a stroke as possible, in order that the distance between the valve and the exhaust ports shall be a maximum. In large engines the inlet valve is always mechanically operated, and since its operation is independent of the piston, its opening can be delayed until the exhaust pressure has fallen below the scavenging pressure, and it can be held open until after the exhaust ports are closed. For this reason it has all the advantages possessed by the bottom scavenging with the delaying valve, and also the additional advantage that, since the exhaust ports are arranged all round the cylinder, ample area can be provided without encroaching too far on the expansion stroke or unduly weakening the cylinder.

The disadvantages are: (1) To give sufficient area the valve, if of the ordinary poppet type, must be very large and heavy, and, since it has to be fully opened and closed in only about 90 degrees movement of the crank, a very powerful spring must be used, involving heavy operating gear and considerable noise, even at low speeds, and rendering high engine speeds impossible. (2) The provision of such a large valve, or in some cases a number of valves,

in the cylinder head weakens this part very seriously, and also prevents efficient water-cooling. To give some idea of the size of valves necessary, let it be assumed that the piston speed of the engine is 900 ft. per minute, or 15 ft. per second, and that the maximum permissible velocity through the valve is to be 150 ft. per second. Then, if the valve be regarded as effectively open during 72 degrees, or equivalent to $\frac{2}{5}$ of the stroke of the engine, it follows that its area must be $\frac{15}{150} \times \frac{5}{2}$ times the area of the piston $= \frac{1}{4}$ of the area of the piston; that is to say, the diameter of the valve port should be equal to half the diameter of the cylinder for a piston speed of only 900 ft. per minute. In practice such large valves are not often fitted, and it has been considered preferable to employ higher gas velocities, in spite of the extra work thrown on the scavenging pumps. In the case of small engines, especially when the scavenging pumps are supplying combustible mixture, it is common practice to dispense with any operating gear, and to fit the valve with a comparatively light spring, so that it opens automatically as soon as the exhaust pressure drops below the scavenging pressure, and closes when the compression pressure exceeds it. To accomplish this, it is desirable that the scavenging pump be so timed in relation to the main piston that the scavenging pressure is removed, and the valve allowed to close, as soon as, or shortly after, the piston covers the exhaust ports. This arrangement gives fairly satisfactory results at low speeds; but poppet valves when operating automatically are liable to be very noisy, and high speeds are rendered impossible owing to their inertia, while any increase in the spring tension increases not only the noise, but also the negative work of the pump. Moreover, such valves are liable, sooner or later, to break, generally under the head, owing to the severe hammering they are subjected to, unless they can be partially cushioned by some form of dash-pot.

Double-piston Scavenging.—The third method of scavenging, shown in fig. 74*c*, is efficient, and has much to recommend it, but it involves the employment of two pistons moving in opposite directions. In this method, one piston uncovers a complete ring of exhaust ports, while the other shortly afterwards uncovers a second ring of inlet ports at the opposite end of the cylinder. To obtain the requisite delay between the opening of the exhaust and inlet ports, either of two methods may be employed. (1) The exhaust ports can be made larger than the inlet. This is open to the objection that too much of the expansion stroke is wasted; however, this is not so serious as in the engine with a single piston, for both

the exhaust and inlet ports can be arranged round the whole circumference of the cylinder, and sufficient area provided, within a comparatively short length. (2) The two pistons can be arranged so as to be out of phase to the extent of about 15 degrees, that is to say, the piston controlling the exhaust ports can have a lead of 15 degrees over the inlet piston. In this case the two sets of ports can be made identical, and the inlet consequently closes 15 degrees after the exhaust, which is a distinct advantage.

When used as a gas-engine it is usual to provide two rows of inlet ports—one for air, which is opened first and closed last, and one for gas, which is only opened when the piston is near the extreme dead centre. In this way a charge of air is first introduced, followed by gas and air, and lastly air alone, so that if diffusion does not take place to any great extent, each piston is protected by a layer of air. It is probable, however, that with such an arrangement diffusion is fairly complete before ignition occurs. From the point of view of governing, the engine is bad, the mixture being too weak and too diffused to ignite regularly on light loads. The use of two pistons has much to recommend it for very large engines apart from the scavenging, but the advantages will be considered later, when dealing with the particular engines in which this system is employed.

U-shaped Cylinder Engine.—The arrangement shown in fig. 74*d* is really a modification of 74*c* in which the cylinder has been bent double in the middle. In this arrangement two pistons reciprocate together in two cylinders which are joined together at the top, forming a common combustion chamber. As in *c*, one piston controls the inlet and the other the exhaust ports, which are the maximum possible distance apart. The two pistons may be out of phase to the extent of from 15 to 20 degrees. The effect is, of course, precisely the same as in case *c*, but the engine is very much more compact, and consequently a long stroke can be employed without unduly increasing the length or height. It has not, however, the advantage possessed by *c* that the reciprocating parts are balanced. When used as a gas-engine, a certain amount of stratification can be obtained by suitable design of the combustion chamber, and very fair governing is effected if the igniter be placed directly over the intake piston. With this arrangement, in order to reduce the clearance space necessary for high compression and at the same time avoid throttling the incoming charge, it is preferable to make the combustion chamber into a kind of curved Venturi tube, as

shown in the diagram. Apart from the question of scavenging, this form of cylinder has other advantages, which will be discussed later.

Scavenge Pumps.—The next point to be considered is the method of supplying the scavenging air to the cylinder. In the case of very large oil-engines of the Diesel type, this is generally accomplished by means of a double-acting reciprocating pump, usually provided with piston valves, operated by an eccentric. Sometimes, however, especially in the case of reversible marine engines, automatic plate valves, such as are commonly employed for modern low-pressure air compressors, are fitted. From the pump the air passes into a receiver of considerable capacity, and thence into the cylinders. A pressure of from 3 to 5 lb. is usually sufficient, and the volume swept by the scavenge-pump piston is generally from 30 to 60 per cent greater than that swept by the working pistons. The actual quantity of air pumped is probably unknown, for it is by no means an easy matter to measure large volumes of air when the flow is pulsating and not continuous. No control is fitted to these pumps, which consequently always discharge the full volume of air irrespective of the load on the engine.

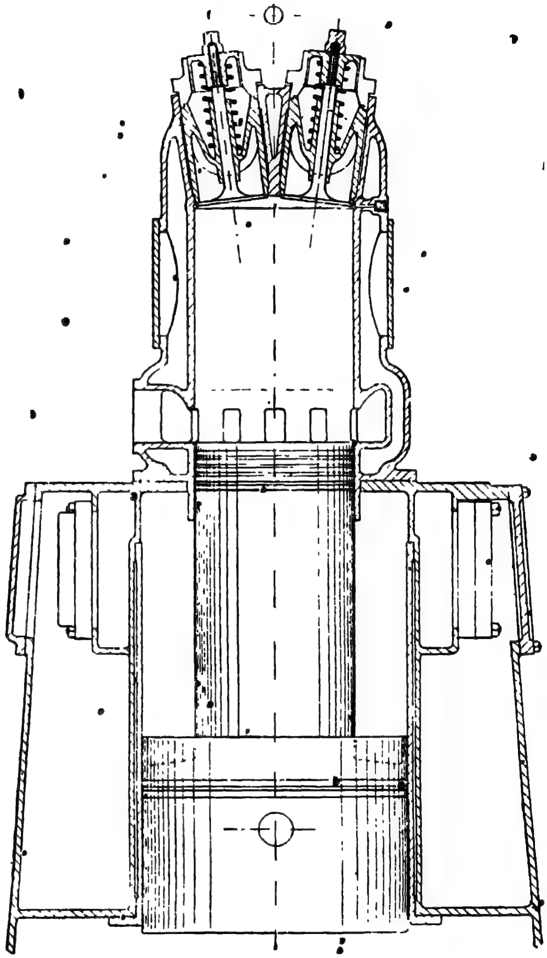


Fig. 77.

In the smaller Diesel engines it is a very common practice to use what is known as a stepped piston, that is to say, the bottom of the piston is enlarged about 50 per cent in diameter, and works in the lower part of the cylinder, which is similarly enlarged, as

shown in fig. 77. The annular space thus formed is used for pumping air into a receiver, light spring automatic valves being used for the suction and discharge. This arrangement has simplicity and neatness in its favour; but since it adds considerably to the reciprocating weight, and moreover spreads the centres of the cylinders and so increases the length of the engine, it is hardly to be recommended. Also the efficiency of the pump is seriously reduced, owing to the high temperature of the piston which tends to heat and expand the air on its entry, and so to reduce the weight of air dealt with. Not only does this lower the efficiency of the pump, but, unless an after-cooler is fitted, unnecessarily hot air is delivered to the working cylinder, a most objectionable feature, in that it both reduces the specific capacity of the cylinder and raises the temperature throughout the cycle.

Base-chamber Compression.—For small engines of the so-called semi-Diesel type it is usual to employ the pressure set up in a closed crankcase for scavenging. Except for extreme simplicity and low first cost, this system has nothing to recommend it, but since it is so widely used it is perhaps worth while to consider it in some detail. In the first place, however carefully the base chamber be designed, a clearance space of at least 150 per cent is inevitable. At the time when the inlet ports are first opened the clearance space will exceed 200 per cent. If now the average back pressure from the exhaust amounts to 3 lb. per square inch during the charging period, the amount of air that will be delivered to the cylinder is about 60 per cent of the swept volume, for the pressure in the crankcase will be only $\frac{300}{200} \times 14{\cdot}7$ lb. $= 22{\cdot}05$ per square inch absolute, or $7{\cdot}35$ lb. per square inch gauge The amount of air delivered will be $\frac{7{\cdot}35 - 3}{7{\cdot}35} = 59$ per cent of the swept volume. It must of course be understood that such a calculation is but a rough approximation, and is only intended to give a general idea. It is based on the assumption (1) that the crankcase is completely refilled during the upward stroke of the piston; (2) that the air is not heated during the suction or compression strokes. In practice the proportion of air actually retained in the cylinder seldom exceeds 35 per cent, for (1) the crankcase is not completely filled owing to the resistance and friction of the inlet valve; (2) the air is very considerably heated on the suction stroke by the hot piston and other parts; (3) the inlet ports offer a very

considerable frictional resistance, especially since bottom scavenging without a "delaying" valve is almost invariably employed.

The result of this extremely imperfect scavenging is that only very low mean pressures can be obtained, and the size and weight of the engine is out of all proportion to the power developed. Moreover, the retention of such a large proportion of highly heated exhaust gases, and the intimate mixing which results from this system of scavenging, tend both to retard combustion and to raise the whole temperature of the cycle—two objectionable features. The use of the crank-chamber as a scavenging-pump is a very serious obstacle to the proper lubrication of the parts within it, especially the connecting-rod big-end bearings. Neither splash nor forced lubrication can satisfactorily be employed, for it is clear that any excess of oil will be carried by the scavenging air into the cylinder. For this reason only the very scantiest lubrication can be provided, and it is probable that if the speed and the mean effective pressure were not already limited by the very poor scavenging, any considerable increase in either would be effectually prevented by the inadequate lubrication.

In the author's opinion it is extremely doubtful whether crank-chamber compression should ever be employed, except in the very smallest engines, for the addition of a separate pump will increase the specific power of the engine by about 100 per cent, certainly without adding more than 50 per cent to the weight or cost. The addition of a separate pump, besides increasing the mean effective pressure, will also permit of higher speeds, better combustion, and of the thorough and efficient lubrication of all the working parts. Owing to the very great clearance space in the crank-chamber, and the consequent low pressure of the charge, it is evident, as has been shown above, that any variation in the exhaust back-pressure, by reacting against the scavenge air, will have an enormous influence upon the mean pressure, and therefore upon the output of the engine. An increase of exhaust back-pressure from nil to $3\frac{1}{2}$ lb. per square inch will reduce the volume delivered from 100 per cent to 50 per cent, other things being equal, and so will halve the output of the engine. Where separate pump cylinders are employed for scavenging, and where the clearances are comparatively small, any variation in the exhaust back-pressure will be automatically met by a corresponding variation in the scavenge air pressure, and the effect upon the power or steadiness of the engine will be but slight. To prevent as far as possible any undue exhaust back-pressure, and

also any pulsations in the exhaust in engines using this system of scavenging, it is usual to fit a very large expansion chamber as close as possible to the exhaust ports; but even with this precaution engines employing crank-chamber scavenging are exceedingly sensitive to variations in the exhaust back-pressure. Care must also be taken to ensure that the exhaust pipes leading from the expansion chamber are short, and that the silencer is as free as possible from obstruction.

So far as the author is aware, very few authentic tests have been made to ascertain the exact proportion of air and exhaust gases present in the cylinder during the compression stroke of a two-cycle engine. Such measurements are not easy to make, because of the great difficulty of measuring the amount of air that enters the cylinders, owing to the pulsations set up by the intermittent opening of the ports and pump valves. If diffusion between the air and exhaust gases within the cylinder were complete, it might be possible to obtain some useful data by taking samples from the cylinder during the compression stroke and submitting them to analysis. Although in some cases, especially in the case of bottom scavenging, there is evidence that diffusion is fairly complete, in other cases there is equally strong evidence to show that very little diffusion occurs; consequently analysis would show widely varying results, depending upon the part of the cylinder from which the sample was taken. Two reliable tests, however, have recently been published, one by Professor Watson on a small Day two-cycle petrol-engine, using bottom scavenging and crank-chamber pumping, and one by Professor Hopkinson on a Fullagar gas-engine, using port scavenging with opposed pistons and a separate pump. The results of these investigations are as follows:—

PROFESSOR WATSON'S TESTS ON A DAY ENGINE

Diameter of cylinder	3·25.
Stroke of piston	3·25.
Swept volume	27 cu. in.
Total volume (from bottom of stroke) ...	34·5 cu in
Volume swept by pump piston	27 cu. in.

R.P.M.	Piston Speed (feet per minute).	Volume delivered by Crankcase (cubic inches per stroke).	Volume lost through Exhaust (cubic inches per stroke)	Volume retained in Cylinder (cubic inches per stroke).	Ratio of Volume retained to Total Volume.
		cu. in.	cu. in	cu in	
600	325	16·7	5·1	11·6	33·6 per cent
900	497·5	17·1	5·7	11·4	33·0 ,,
1200	650	14·6	3·5	11·1	32·2 ,,
1500	812·5	12·6	2·4	10·2	29·3 ,,

The tests given above represent the very best results which Professor Watson was able to obtain, after modifying the ports in such a manner as to obtain the highest possible volumetric efficiency in the crankcase.

Professor Hopkinson's tests on a Fullagar opposed piston engine of 500 B.H.P., running at 250 R.P.M., gave the following results:—

Diameter of cylinder	12 in.
Stroke of each piston	18 in.
Swept volume	2·36 cu. ft.
Total volume	2·75 cu. ft.

R.P.M.	Piston Speed.	Volume delivered by Blower.	Volume retained	Ratio of Volume retained to Total Volume.
		cu. ft.	cu. ft.	
250	750	2·70	1·78	65 per cent
250 (half load)	750	1·42	1·20	43·5 "

In the case of the Fullagar engine the scavenge air was delivered to the ports by a separate blower, the measurement being made on the intake side of the blower. If, as may be supposed, these results are accurate, the percentage of air lost to the exhaust clearly points to the fact that diffusion has taken place to such a considerable extent that the inlet and exhaust gases are almost completely mixed before the closing of the ports. It should be pointed out that in this particular engine there is no lead on the piston which uncovers the exhaust ports, hence these ports are closed considerably after the inlet. No doubt, from the point of view of scavenging, a considerable improvement could be effected by giving the exhaust piston a lead of 15 to 20 degrees, and reducing the height of the exhaust ports correspondingly. It will be observed, however, that the percentage of air retained in the cylinder of the Fullagar engine is practically double that retained in the Day cylinder, using crank-chamber displacement and running at the same piston speed.

For the purpose of considering both engines on the same basis, the author has referred the percentage volume of air retained to the total volume of the cylinder, when the pistons are at their extreme out centre. With two-cycle engines it is always difficult to decide where the effective stroke ends. It is, certainly at some point considerably before the extreme dead centre, though exactly how much before must depend upon the design of the ports and the speed at which the engine is running. In practice, when considering the

mean pressure of two-cycle engines, it is usual to refer it to the full stroke, and this will be done throughout this book. In the four-cycle engine it is usual to commence opening the exhaust valve about 10 per cent before the end of the stroke, but since this valve is relatively small, and the opening relatively slow, the pressure drop before the end of the stroke amounts to very little. In two-cycle engines, on the other hand, the exhaust ports usually commence to open from 20 to 25 per cent before the end of the stroke, and since their area is large and the rapidity of opening great, the pressure

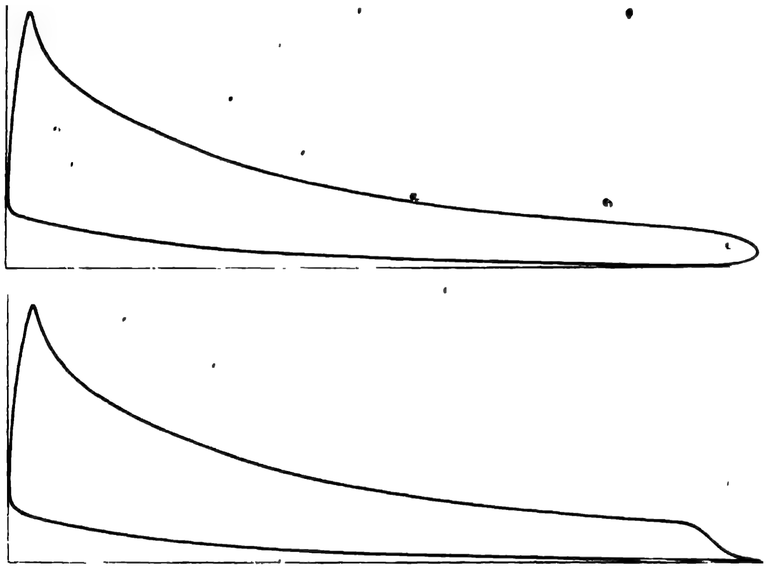


Fig. 78

in the cylinder drops abruptly, and no further work is done on the piston during the last 10 per cent or so of the stroke. This difference is clearly shown in fig. 78, where two actual diagrams, one from a two-cycle and one from a four-cycle engine, are reduced to the same scale and superimposed. Unfortunately, however, the diagram taken from the two-cycle engine shows a loss far below the average, the engine being a very slow-speed type with exhaust ports all round the circumference. In comparing the performance of two- and four-cycle engines this feature should be borne in mind, and it must be remembered that the loss of 10 or 15 per cent of the stroke is the forfeit that a two-cycle engine must pay in order to obtain an impulse every revolution.

Scavenging of Gas-engines.—When the two-cycle system is employed for explosion engines, the difficulty of efficient scavenging

is considerably increased, for not only is it necessary to expel as much as possible of the exhaust gas, but this must be done without the loss of any combustible mixture. Any loss of air through the exhaust is serious in that it increases the work thrown upon the scavenging pump, but this is insignificant compared to the loss of efficiency which results if any of the fuel is allowed to escape. Attempts are made to ensure against the loss of fuel in two ways: (1) By scavenging first with pure air and then adding the gas, or a rich mixture of gas and air, at a later period. In this case it is assumed that whatever loss occurs is of air and not gas, since the air, having entered first, is the first to reach the exhaust ports. (2) By so designing the entry port or valve, and the whole of the combustion space, that the fresh charge enters at low velocity, generally in the form of a gradually expanding cone, the object being to prevent diffusion and encourage stratification. If there be no diffusion, it is evident that there will be no loss of fresh charge through the exhaust ports, until the whole of the exhaust gases have been expelled. With this system, which in the author's opinion is the better one for small engines, no separate air-scavenging charge is required, consequently only one pump is needed, supplying a mixture of gas and air.

After numerous tests, the author has found that, for all-round work in the case of small engines, there is very little to choose in overall efficiency between the two systems, for the extra power required to drive two separate pumps neutralizes the advantage that the air-scavenging system may possess in the way of immunity from fuel loss through the exhaust ports. In this, as in all other questions of design, consideration must be given to the size and the particular conditions under which the engine will be required to run. For large engines the air-scavenging is probably to be preferred, because: (1) With very large pumps delivering combustible mixture there is always a risk that an explosion in the pump may cause considerable damage. (2) In large engines where the rotational speed is comparatively low, the frictional and fluid losses due to the two pumps are smaller in comparison; their losses in a well-designed modern gas-engine should not together exceed 10 per cent of the total indicated horse-power.

The objections to the air-scavenging system are:

1. The use of two independent pumps is costly, complicated, and involves a considerable addition to the mechanical friction.

2. In spite of the use of separate pumps, if the gas and air both

enter through the same valve or ports, there is a danger of the gas being driven back into the air passages and pump, or vice versa, in which case the advantage of air-scavenging is lost. It is by no means easy to arrange for separate valves or ports for the air and gas, nor is even that a certain remedy if both are in communication with the cylinder at the same time. This difficulty can be overcome to a certain extent by the employment of a secondary air-scavenge, that is to say, by the admission of a second charge of pure air following the charge of gas, in order to clean out the ports and passages.

3. The difficulty of governing. If the charge is to be homogeneous, it is necessary that there shall be complete diffusion of the gas and air; and if they are admitted separately, such diffusion must take place within the cylinder. This is the condition necessary for full-load running. If it be required to run on light loads, and the proportion of gas is considerably reduced, such diffusion will result in the formation of a mixture too weak to ignite. If it were possible so to design the engine that the gas and air were thoroughly diffused on full loads and stratified on light loads, the necessary conditions would be fulfilled, but it is not easy to see how this could be accomplished.

By employing a single pump delivering a homogeneous mixture of gas and air, and aiming at complete stratification of the combustible mixture and exhaust gases, governing is rendered fairly simple by merely regulating the quantity of combustible mixture admitted at each stroke, a small proportion only of the exhaust gases being expelled on light loads. At the same time a very much cheaper and simpler engine can be produced, and the friction and fluid losses can be reduced to a minimum. The objections to this system are:

1. The danger referred to before of a fire-back into the pump cylinder. In comparatively small engines it is easy to make the pump mechanism strong enough to withstand this, for, since the compression realized in the pump cylinder seldom exceeds 3 or 4 lb. per square inch, the maximum pressure attained in the event of an explosion in the pump is not very high.

2. Since complete stratification is not obtainable, a certain proportion of the combustible gases does mix with the exhaust and pass out of the exhaust ports. This loss is very small when running on light loads, but may become serious when the proportion of fresh charge exceeds about 50 per cent of the total cylinder volume.

Comparison with Four-cycle Engine.—It is popularly supposed that the two-cycle engine is vastly inferior to the four-cycle in thermal efficiency, and that the fuel consumption is very much greater. However true this may have been in the past, the difference at the present time between the best examples of each type is not so very great, being usually only about 10 per cent in favour of the four-cycle. This may be attributed to three main causes:

1. The lower mechanical efficiency of the two-cycle engine, due to the separate scavenging-pump with its greater friction and fluid losses.
2. The fact that after combustion the gases are not expanded to the same volume as before.
3. The loss of unburnt fuel through the exhaust in the case of explosion engines only.

The first of these causes is quite obvious, for it follows that the extra piston of the scavenging-pump, together with its operating gear, must add somewhat to the friction, and that, owing to the short time available for charging, the gas velocities must be higher, with a consequent increase of fluid friction. Moreover, the valves of the scavenging-pump itself will add still further to the fluid losses. When, however, all these conditions are taken into consideration, the difference in mechanical efficiency amounts in large engines to only some 6 per cent in favour of the four-cycle engine. In cases where the crankcase is employed for scavenging, the extra friction loss due to a separate pump is avoided, but the fluid losses are increased proportionately, owing to the fact that the air or gases are compressed in the base-chamber to a considerably higher pressure than is necessary, and then released without doing any useful work when the inlet ports are opened. An average of a great number of tests shows that the mechanical efficiency of a good modern two-cycle gas-engine may be taken as about 80 to 82 per cent for all sizes when the piston speed is about 750 ft. per minute. Larger sizes do not show better results, because separate air-scavenging is generally employed. The mechanical efficiency of a four-cycle engine generally ranges from 86 per cent in small sizes to 88 per cent in large engines. In the case of Diesel oil-engines, owing to the power required to drive the high-pressure air-compressor and the heavier reciprocating parts, the mechanical efficiency of both is some 12 per cent lower, and the difference is consequently even less.

The second cause is less obvious, for the point at which compression may be said to commence is not very definite. As a general rule, the pressure in the cylinder is about 17 lb. per square inch absolute at the moment when the inlet port or valve is closed, which may be assumed to occur when the piston has travelled over 20 per cent of the compression stroke. Then, assuming adiabatic compression, the point in the stroke at which the pressure would be atmospheric may be found as follows:—

$$x^{1\cdot 4} \times 14\cdot 7 = 17,$$
$$x^{1\cdot 4} = \frac{17}{14\cdot 7}$$
$$= 1\cdot 16,$$
$$\text{whence } x = 1\cdot 175.$$

That is, the compression commences 17·5 per cent before the closing of the inlet valve, or 2·5 per cent after the out centre; but, as has been shown before, owing to the large area and the rapid opening of the exhaust ports, the expansion stroke may be considered as complete when the piston is 10 per cent from the end of its stroke. If, therefore, 17 lb. per square inch be taken as the pressure in the cylinder when the ports are closed, and it is about the average figure, then the gases are expanded to only 92·5 per cent of their original volume. In many cases the expansion ratio is even less than this. The effect of this, of course, is to increase the pressure and temperature at the point of release and lower the efficiency, owing to the greater amount of heat rejected to the exhaust. The loss due to this cause is not large, but it must be taken into consideration when accounting for the lower efficiency of the two-cycle engine.

The third cause, that due to loss of unburnt fuel through the exhaust, is only of importance, among well-designed engines, in the case of those which compress combustible mixture; it depends very largely upon the design of the cylinder, the exact timing of the valves or ports, and the length and size of the exhaust pipes. With engines using bottom scavenging, no delaying valves, and only a deflector on the piston-head to prevent short-circuiting, this loss is certainly large, in some cases exceeding 30 per cent. Such a system of charging, however, cannot be too strongly condemned for engines which scavenge with combustible mixture. When scavenging is effected through a valve in the cylinder, by so designing the cylinder as to encourage stratification it is possible

to reduce this loss to a very small percentage. With opposed pistons, owing to the diffusion which takes place, the percentage is larger, but still need not be great.

But for the deficiencies just mentioned the thermal efficiency of a two-cycle engine should be higher than that of a four-cycle employing quantitative governing when running on light loads; for, since the weight of gas in the cylinder is always approximately the same, a comparatively high mean pressure can be obtained with a small rise of temperature, and consequently the losses due both to the increasing specific heat of the gases at high temperatures and also to radiation are greatly reduced. In other words, a two-cycle engine employs qualitative governing, with all the advantages which that system bestows.

CHAPTER XVIII

EXAMPLES OF LARGE TWO-CYCLE GAS-ENGINES

The Oechelhauser gas-engine is probably the best-known example of the type in which the system of secondary pure-air scavenging is employed. In this engine, as will be seen from the sectional drawing in fig. 79, two pistons are employed in one long cylinder. One piston uncovers the exhaust ports, and the other two rows of inlet

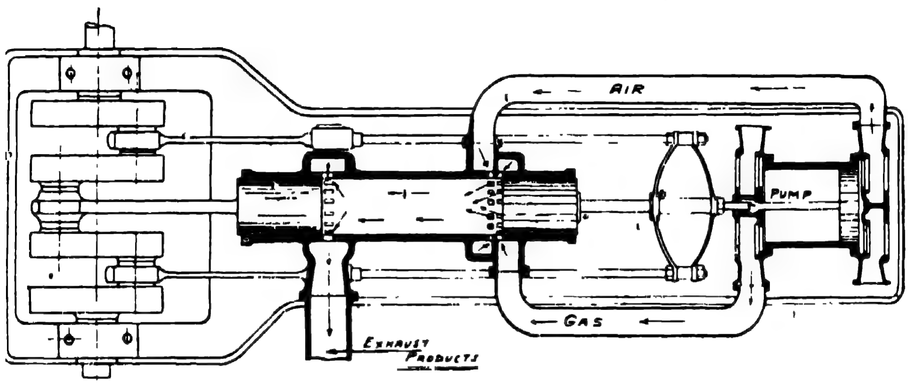


Fig. 79.—Diagram illustrating Principle of the Oechelhauser Engine

ports. The first row to be uncovered, and the last to be closed, admits pure air, while gas is admitted through the second series. In this way, primary and secondary pure-air scavenging is effected in a very simple manner, and with little risk of any gas passing into the scavenging air. It is also claimed that each piston is protected by a layer of pure air, but, as has been pointed out previously, there is probably far too much diffusion for this to be of any consequence. The use of two pistons travelling in opposite directions has certain very marked advantages both from the thermodynamic and mechanical point of view. Considering first the thermodynamic advantages. (1) A very long effective stroke can be employed in conjunction with a high rotative speed so that the expansion is much more rapid than when a single piston is used, consequently the heat loss during the expansion stroke is reduced. (2) The combustion chamber presents the

minimum possible surface; for, owing to the long stroke, the layer of gases at the end of compression is very thick, while the exposed surfaces consist only of a portion of the cylinder liner and the two piston-heads. There are no pockets or recesses which can check the free movement of the gases during combustion, so that the combustion of the whole mass of the gas is very rapid. (3) The absence of any uncooled valves or considerable thickness of metal at any one point, which may become overheated, permits of the use of a higher compression, without risk of pre-ignition, than is possible with other designs.

The mechanical advantages are: (1) The two pistons moving in opposite directions give a nearly perfect balance of the moving parts, thus permitting of high rotative speeds. (2) The cylinder-head, always the most troublesome part in large engines, is eliminated, and replaced by a second piston. (3) Since the crank is pushed by one piston and pulled by the other, the bearings, bedplate, and framing are relieved from all stresses under normal conditions, and they can therefore be made very light. (4) Since the cylinder is a plain barrel of uniform thickness it is easy to cast, and is less liable to internal stresses, due to contraction, such as occur in the cylinder-heads of single-piston engines; for in such engines the casting becomes complicated, owing to the valve ports, passages, &c., and is seriously weakened both by internal cooling strains and by temperature differences, due to the unequal thickness of the metal. Much is claimed for this feature, but in practice the advantage is not so great as might at first sight appear, for it is necessary to provide bosses, and to pierce the liner in several places for the igniters and starting valves, and these necessarily interfere with its free expansion, and are a source of local weakness.

The principal objections to this class of engine are: (1) Owing to the necessity for a three-throw crank and three connecting-rods, cross-heads, slides, &c., required for each single-cylinder unit, the cost of manufacture per effective horse-power is very high, and the engine has had difficulty in competing with the double-acting two-cycle engine on this account. (2) Owing to the long return connecting-rods, necessary to connect the outer pistons with the crankshaft, the length or height of the engine is very considerable, and the weight of the reciprocating masses is great. Against this latter objection, however, it must be remembered that they are well balanced, and external cross-heads are provided which greatly reduce the piston friction.

Actual tests on Oechelhauser engines have given the following excellent results:—

Test I: 500 horse-power single-cylinder Oechelhauser gas-engine, tested by Professor E. Meyer in the Borsig Works at Tegel, near Berlin. This engine consumed coke-oven gas containing between 40 and 50 per cent by volume of hydrogen (H_2), and having a lower heating value of 350 to 433 B.T.U. per cubic foot.

The following are the leading dimensions of this engine:—

Power cylinder	Diameter, 26·6 in. Stroke front piston, 37·5 in. Stroke back piston, 37·31 in.	Combined stroke, 74·31 in. Swept volume = 20 cu. ft. (approx.).
Air-scavenging pump (double acting) ...	Bore, 44·85 in. Stroke, 19·7 in.	Swept volume = 30·0 cu. ft. (approx.).
Gas-pump (double acting)	Bore, 23·2 in. Stroke, 19·7 in.	Swept volume = 8·0 cu. ft.

The ratio between the volume swept by the power pistons and the air-scavenge pistons is, therefore, 1 : 1·5, a very high ratio. If the capacity of the gas-pump be added to that of the air-scavenge, the ratio then becomes 1 : 1·9.

In this particular instance the engine was employed for driving a blowing-tub, and the mechanical friction includes that due to the piston and stuffing-boxes of the blower, for the brake horse-power of the engine is deduced from the power indicated in the blowing cylinder.

NUMBER OF TESTS, AUGUST, 1904.

	I	II	III	IV	V	VI	VII	VIII	IX
R.P.M.	110·6	113·6	112·9	108·3	96·4	84·5	86·8	86·1	68·4
Piston speed (feet per minute)	690	708	702	675	600	527	532	530	426
Combined piston speed as affecting expansion	1380	1416	1404	1350	1200	1054	1064	1060	852
M.E.P. (indicated) ...	74·5	60·5	57·1	41·2	32·6	70·1	53·4	52·0	57·1
η_p	53·3	39·7	36·7	23·3	13·4	51·6	37·7	37·3	40·8
I.H.P.	865	723	676	468	330	622	486	470	410
B.H.P. (indicated in blowing cylinder) ..	619	476	433	269	144	457	343	338	293
M.E.P. air-pump ...	7·30	6·70	6·80	5·00	5·85	6·93	4·70	4·60	4·10
I.H.P. air-pump ...	104·0	98·6	98·7	72·5	71·5	71·7	52·6	52·1	34·8
M.E.P. gas-pump ...	3·11	3·45	3·52	3·56	3·69	3·53	3·15	3·34	3·28
I.H.P. gas-pump ..	7·25	8·25	8·37	8·14	7·51	6·29	5·76	6·06	4·73
Total pump I.H.P. ...	111·2	106·8	107·0	80·6	79·0	78·0	58·3	58·6	39·5
Fluid losses, per cent	12·8	14·8	15·8	17·4	23·8	12·5	12·0	12·5	9·64
Friction losses, per cent	15·5	19·6	20·0	25·3	32·4	14·0	17·4	15·7	18·9
Mechanical efficiency	71·7	65·4	64·2	57·3	43·8	73·5	70·6	71·8	71·5
Thermal efficiency (indicated)	37·8	39·0	39·0	35·8	26·3	32·3	36·3	38·0	37·1
Brake thermal efficiency (per cent) ...	27·3	25·5	25·0	20·5	11·5	23·7	25·7	27·3	26·6

The thermal efficiency, in terms of the indicated horse-power in the power cylinder, as shown here, is remarkably good, and is equal to the best results ever obtained from a modern four-cycle engine. But when it is reduced to the B.H.P. the very poor mechanical efficiency rather spoils the performance. It is evident, when analysing these results, that the fluid losses are altogether too high, due partly to the excessive capacity of the air-pump, which is evidently delivering too great a volume of air for the pump valves and inlet ports to pass without serious back-pressure. This is clearly indicated by examining Tests III and IX. The mean pressure is the same in both cases, but the piston speed, and hence the gas velocities, are reduced in IX, while the fluid loss drops from 15·8 per cent to 9·64 per cent. The purely mechanical friction has dropped from 20 per cent to 18·9 per cent, a very small difference.

The admission of this very large volume of air probably improves the indicated thermal efficiency slightly, by lowering the initial temperature of the working fluid, but any gain from this source is more than counterbalanced by the extra fluid and friction losses. With modern two-cycle engines the fluid loss due to scavenging should not greatly exceed 5 per cent, and in this case it is probable that better brake thermal efficiencies would have been obtained had a smaller air-pump been used, and had the exhaust piston been given a slight lead over the inlet, thus permitting of the use of larger inlet ports. This engine affords a very good example of the danger of concentrating too much on the thermodynamic, to the neglect of the mechanical efficiency. Suppose that in Test III less air had been delivered, and that the fluid loss had been reduced to 5 per cent at the expense of 1 per cent loss of thermal efficiency, then the brake thermal efficiency would become 28·5 per cent as against 25 per cent—a very substantial improvement.

In a later test on the same engine, Professor Meyer obtained an indicated thermal efficiency of 39·8 per cent and a brake thermal efficiency of 28·8 per cent; certainly a remarkable result, but the author is not aware what modifications, if any, were made to the engine between these two sets of tests.

Such high indicated thermal efficiencies as these show very clearly that the loss of unburnt fuel through the exhaust ports must be almost negligible.

The manufacture of Oechelhauser engines has been undertaken in Great Britain by Messrs. William Beardmore & Co., Ltd., of Glasgow, who have redesigned and considerably improved upon the

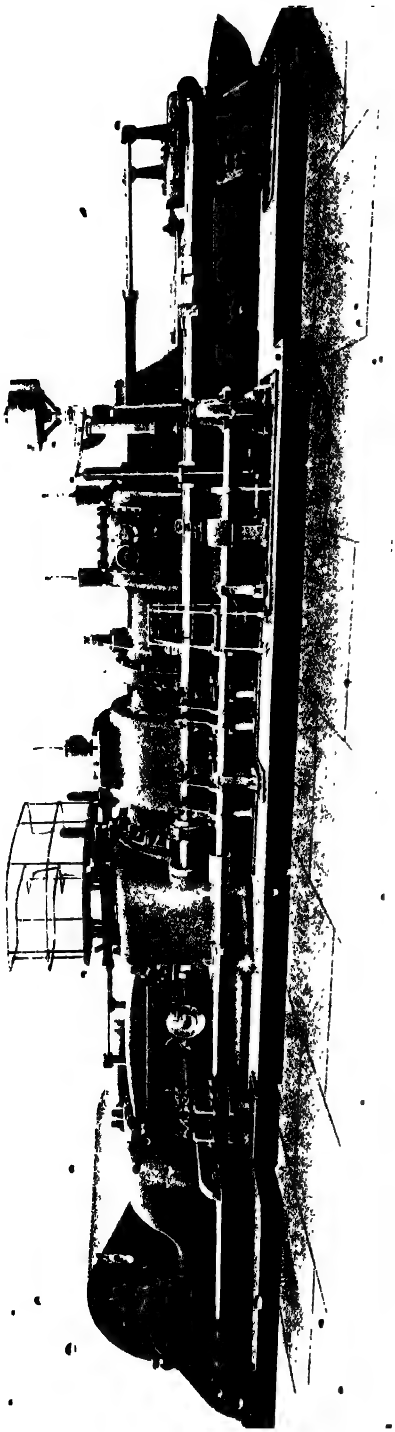


Fig. 80.—1500-B.H.P. Single-cylinder Rolling-mill Engine by Messrs. W. Beardmore & Co., Ltd.

German models. Fig. 80 shows a photograph of a 1500-horse-power single-cylinder rolling-mill engine. Messrs. Beardmore have greatly reduced the friction losses by lightening the reciprocating parts, and at the same time they have entirely redesigned the air-scavenging pump, and have succeeded in reducing the fluid losses to below 7 per cent at full load and normal speed. No very marked improvement has been made in the brake thermal efficiency, but this is probably due to the fact that the engines, as now built, have a very much shorter stroke, necessitated by the strenuous competition which left no place for the earlier costly and bulky long-stroke engines. Moreover, the author is not aware of any instance in which Messrs. Beardmore's engines are using coke-oven gas, which, owing to its richness and rapid inflammability, would tend to show a higher thermal efficiency. With producer gas, Messrs. Beardmore guarantee that the brake thermal efficiency will exceed 25·3 per cent, which compares

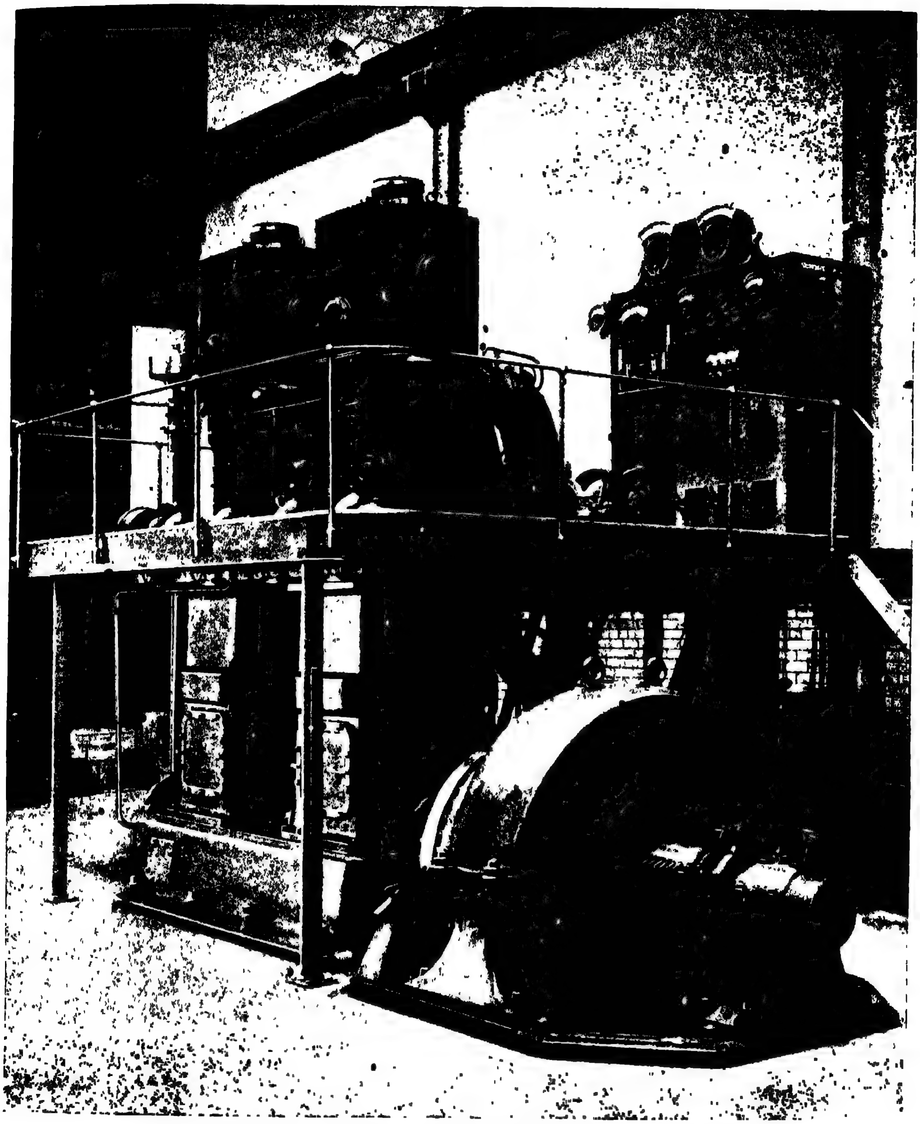


Fig. 81.—Fullagar Engine. Bore, 12 inches; stroke, 18 inches

quite favourably with the guarantees given for modern four-cycle engines.

The Fullagar Engine.—The Fullagar engine, a photo and section of which are shown in figs. 81 and 82[1], is a very modern production, and is in effect a modification of the Oechelhauser engine, in that it employs two opposed pistons in each cylinder. But the

[1] From a paper read by Mr. Fullagar before the North-East Coast Institution of Engineers in 1921.

long return connecting-rods, and the necessity for the use of a three-throw crankshaft, have been done away with by arranging two cylinders side by side, and coupling the lower piston of one to the upper piston of the other, and vice versa, by means of light diagonal steel tie-rods. If the stroke is fairly long in relation to the bore, the angularity of these ties is not excessive, and is, in fact, considerably less than the maximum angularity of the connecting-rods. The thrust due to the diagonal ties is taken by outside cross-heads, which can be kept cool and well lubricated, so that the friction loss due to this cause should be very small. The whole design is attractive in that it is at once simpler, cheaper, and more compact than the Oechelhauser engine. Only the lower pistons are connected to the crankshaft by single connecting-rods, so that a plain two-throw crank with cranks at 180 degrees suffices for both cylinders and all four pistons.

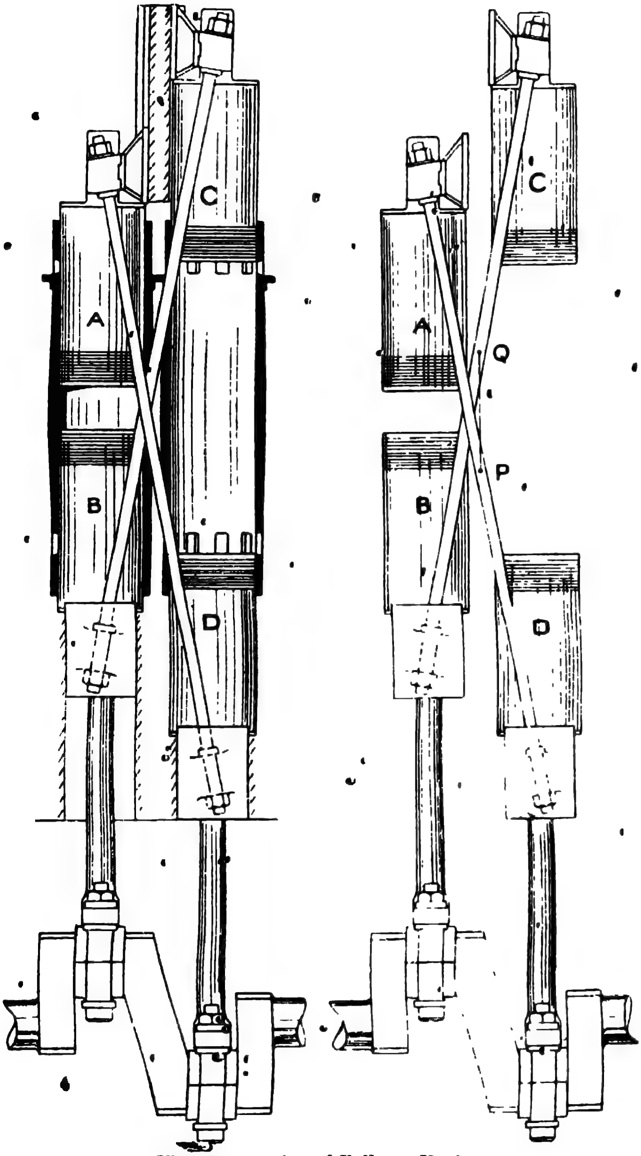


Fig. 82.—Section of Fullagar Engine

In the only example about which the author has any detailed information, air-scavenging alone is employed, the air being supplied

by a rotary fan or blower. As explained previously when discussing scavenging efficiency, the volume of air delivered to the inlet port of this engine is very little in excess of the swept volume of the cylinder, yet the Fullagar engine can obtain as high a mean pressure as the Oechelhauser, indicating that the 50 per cent excess of air supplied to the latter is unnecessary. Gas is supplied to the engine by means of a small reciprocating pump, and is admitted through a valve direct into the combustion chamber, not through ports as in the Oechelhauser. The timing of this valve is such that the gas is driven into the cylinder immediately after the closing of the exhaust ports, and before the compression in the cylinder offers too high a resistance. This method, though excellent, is only practicable with very rich gas, in which only a small volume is required. It would not be possible to admit a large volume of poorer gas into the cylinder after the closing of the exhaust ports, without absorbing a very considerable amount of power in the gas-pump, or introducing so large a valve port in the wall of the cylinder as seriously to weaken the structure of the cylinder itself.

Tests made on this engine by Professor Hopkinson showed an indicated thermal efficiency of 37 per cent, and a brake thermal efficiency of slightly over 30 per cent when running on town's gas. The mechanical efficiency was found to be about 81 per cent, but Professor Hopkinson states in his report that an even higher efficiency would have been obtained had the scavenging been effected by a more efficient form of blower.

The Korting Engine.—The Korting double-acting engine, originally introduced by Korting Brothers, of Kortingsdorf, Hanover, is modelled on the early Clerk engine, and has met with the greatest measure of success of any large two-cycle gas-engine up to the present time. Great numbers of these engines have been built in Germany by various licencees. The engine is no longer manufactured in Great Britain. One of the largest installations of Korting engines is that at Lackawanna Steel Works, Buffalo, amounting in all to 40,000 B.H.P. The installation was manufactured and put up by the De La Vergne Machine Company of New York. Figs. 83 and 84 show a typical single-cylinder Korting engine.

Dealing first with the general features of the design, it will be seen from the photograph, fig. 83, and the sectional drawing, fig. 84, that a double-acting cylinder is employed, with charging or scavenging-valves at each end, and the exhaust ports all round the cylinder

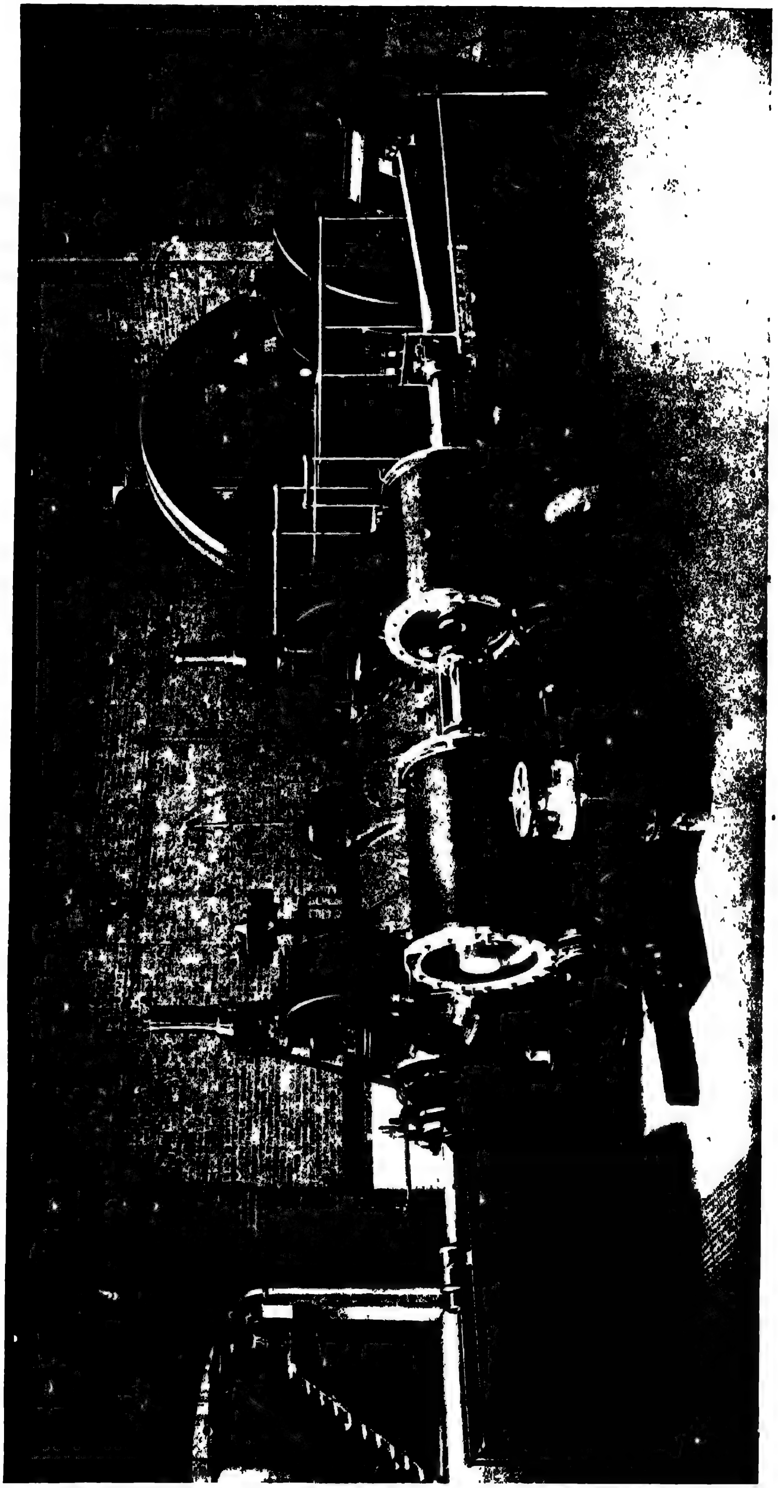


Fig. 83.—The Korting Engine

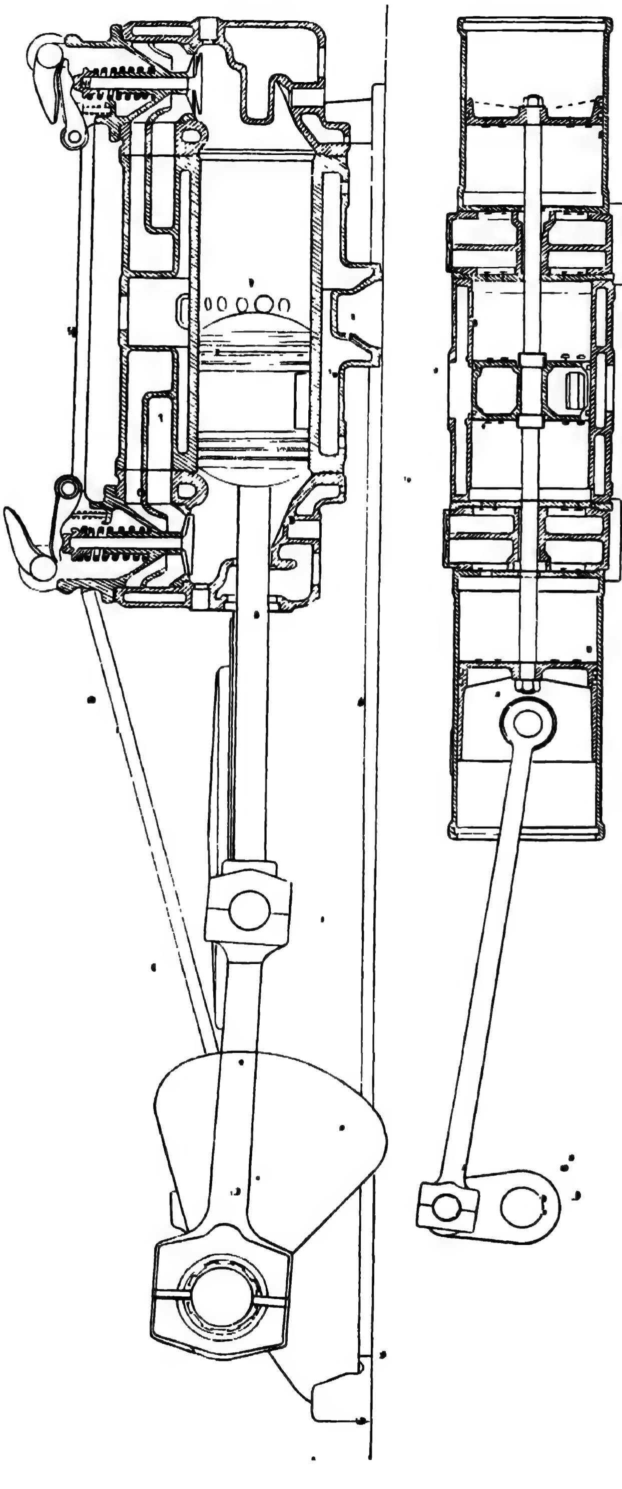


Fig. 84.—Section of the Korting Engine

in the middle. A long piston is fitted, its length being about 20 per cent less than the stroke of the engine, so that the exhaust ports are uncovered to each end of the cylinder during the last 20 per cent of each stroke. The two scavenging-pumps, one for gas and one for air, are placed in tandem at the side of the cylinder, and driven from a separate crank, having a lead of about 110 degrees over the main crank. No receivers are employed, the pumps delivering direct into the cylinder. In this case it is necessary that their delivery stroke shall be so timed in relation to the main piston as to reduce the fluid losses to a minimum. By thus dispensing with the use of receivers it is obvious that the pump losses can be considerably reduced. In order to give a lead to the air, the suction valve of the gas-pump is kept open in the German design during the first 40 to 50 per cent of the pump stroke when on full load, and for a proportionately longer period on light loads, so that the gas is simply returned to the mains until the closing of the suction valve. It is then delivered, along with the air, to the main inlet valve of the power cylinder.

In order to prevent a mixture of gas and air being formed in the passages leading to the main inlet valve, the two fluids are kept separate as far as possible; separate delivery pipes are employed, and even the valve seating has separate passages for the entry of gas and air. The danger of a mixture forming in the passages behind the valve is twofold.

1. Any mixture so formed will enter the cylinder ahead of the air-scavenge charge, and therefore is liable to be driven out through the exhaust ports.

2. In the event of combustion not being complete when the valve first opens, the mixture in the passages may be ignited by flame lingering in the cylinder. This in itself is of little consequence, but it is liable to ignite the whole of the gas and air of the next charge before it can enter the cylinder, thus missing one stroke and fouling a portion of the air for the next stroke. In spite of the precautions taken to prevent it, firing in the gas and air passages does occasionally occur, and indicates that a certain amount of mixture must form behind the valve at the end of the charging stroke. Whether it betrays its presence by igniting or not, the larger part of this mixture will be driven through the exhaust ports unburnt. In this respect the secondary scavenging arrangement of the Oechelhauser engine would appear to be superior.

In spite of the large number of Korting engines in use, no

details of complete tests, such as Professor Meyer's on the Oechelhauser engine, appear to be available. M. Mathot, however, in his book on the *Construction and Working of Internal-combustion Engines*, quotes the following tests carried out on a German Korting engine in 1904, the same year as Professor Meyer's tests on the Oechelhauser engine referred to above. The engine in this case was a single-cylinder unit of 700 B.H.P., running at 80 R.P.M.; the bore and stroke of the cylinder were 31·0 and 55·12 in. respectively. The dimensions of the gas- and air-pumps are not given, but the combined capacity of the two can be calculated back from the M.E.P., the stroke, and the I.H.P., all of which are recorded. Calculated in this way, the bore and stroke are approximately 38 in. by 42 in., and the total swept volume 45 cu. ft. Such an approximation is necessarily rather rough, because the M.E.P. of the two is not exactly the same, nor is the proportion known; but since producer gas was used it is probably approximately in the ratio of 2 : 1.

The swept volume of the working cylinder per stroke is 22·5 cu. ft., and that of the combined gas and air cylinders 45 cu. ft.; but it must be remembered that about 50 per cent of the gas is returned to the suction, so that the actual volume would probably be not more than 37·5 cu. ft., a ratio of 1 : 1·67.

R.P.M.	80.
Piston speed (feet per minute)	732.
M.E.P. (mean of both ends)	56·8.
ηp (pounds per square inch)	45.
I.H.P....	890.
B.H.P.	703·8.
I.H.P. of pumps	91·9.
M.E.P. air-pump	2·756.
M.E.P. gas-pump	3·138.
Fluid losses	10·3 per cent.
Friction losses	10·6 ,,
Mechanical efficiency ...	79 1 ,,
Thermal efficiency (B.H.P.)..	28·6 ,,
Thermal efficiency (I.H.P.) ..	36·2 ,,
Fuel used	Anthracite, 0·78 lb. per B.H.P. hour (14,200 B.T.U.s per pound).

The figures given for the thermal efficiency are based on the assumption that the producer efficiency is 80 per cent. It is very interesting to compare these three results, all from two-cycle engines of approximately the same power—two of which, however, are old-

fashioned and one modern—with a test given also by M. Mathot on a two-cylinder four-cycle double-acting gas-engine of 600 B.H.P., built by Messrs. Ehrhardt & Sehmer, and tested in 1906. This engine had cylinders of 24·4-in. bore by 29·52-in. stroke, and developed 600 B.H.P. at 150 R.P.M. The volume swept by each piston is 7·9 cu. ft., or, for the two pistons, 15·8 cu. ft. per stroke.

R.P.M.	150.
Piston speed	736 ft. per minute.
M.E.P. (mean of four ends)	74·0 lb. per square inch.
ηp	61·5 " "
I.H.P.	723.
B.H.P.	600.
Mechanical efficiency	83 per cent.
Thermal efficiency (B.H.P.)	31 "
Thermal efficiency (I.H.P.)	37·3 "
Fuel used	Coke oven gas, 460 B.T.U.s per pound.

The mean brake thermal efficiency of the three two-cycle engines is 29 per cent, while the brake thermal efficiency of the four-cycle, which may be regarded as a good average example of engines of that size, is 31 per cent, a superiority of only 7 per cent in favour of the four-cycle engine. Comparing the indicated thermal efficiencies, the mean of the three two-cycle engines is 37·7 per cent, while the four-cycle gives 37·3 per cent, a superiority of 1 per cent in favour of the two-cycle engine.

A 2000 Horse-power Cylinder.—In fig. 85 are shown a plan and sectional elevation of a 2000 horse-power single-cylinder engine by the Siegener Maschinenbau, which may be regarded as typical of German design. It will be observed that the cylinder is made in three parts. (1) The water-jacket, which contains also the exhaust belt surrounding the exhaust ports, and which is a comparatively simple iron casting; and (2) and (3), two hard cast-iron liners pressed in from either end of the cylinder so that they almost meet in the middle, a small space being allowed to permit of expansion. The exhaust ports are cut half into each liner. The cylinder covers, which contain the whole of the combustion space, are somewhat complicated castings, and, being necessarily of considerable thickness, are liable to internal strains, due both to contraction in the mould and to the considerable differences of temperature between the inner and outer surfaces of the walls. It will be observed that ribs are formed on the outside of the inner walls

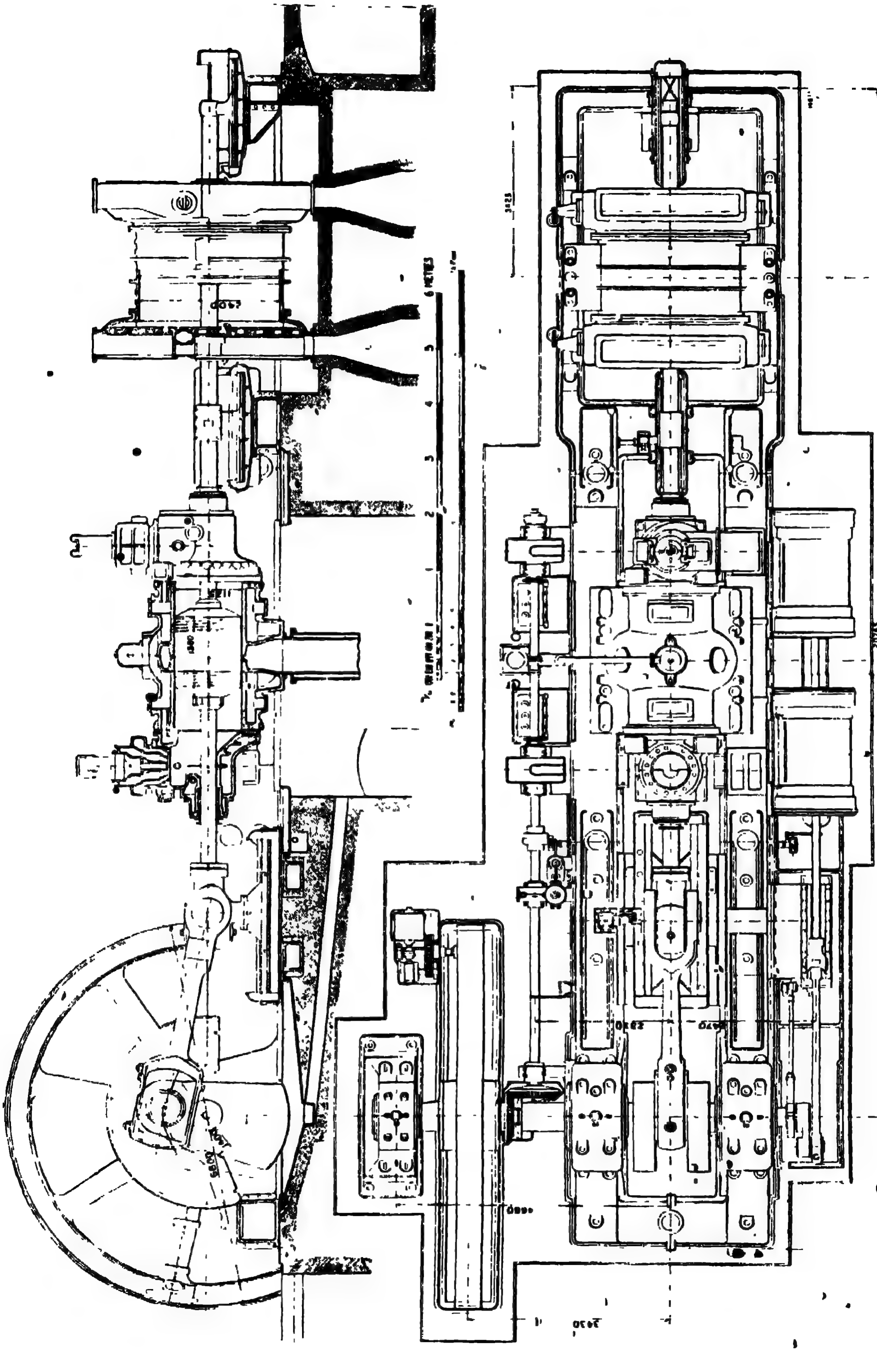


Fig. 85.—2000 Horse-power Single-cylinder Gas Blowing Engine by the Siegener Maschinenbau

in order to increase the cooling surface, and that every effort has been made to allow of expansion of the inner shell without cracking

the water-jacket, one of the commonest sources of failure in engines of this type.

The contours of the combustion chamber are such that the air and gases, after passing through the valve, spread out in the form of a cone, their velocity being reduced as the diameter increases. By this means, diffusion between the air and the exhaust products is reduced to a minimum.

The inlet valve is of enormous diameter, 19·6 in., and has a lift of 3·14 in., giving an effective area of 193 sq. in. when fully opened. This large valve has to be fully opened and closed while the crankshaft passes through about 90 degrees. When running at 90 R.P.M., the engine's maximum speed, the time available for this operation is only one-sixth of a second. The operation of so large a valve and at such a high speed necessitates extreme care in the design of the valve-operating mechanism, which in this case consists of a side shaft driven from the main crankshaft by

Fig. 86.—Section of Korting Piston and Rod

bevel gearing. This shaft is in the form of a two-throw crankshaft with cranks at 180 degrees, and to ensure rigidity a bearing is provided on either side of each crank-web. A connecting-rod connects each crank-pin with the rolling lever or "crocodile jaw" mechanism above the valve. For closing these very large valves springs alone are not relied upon, but are supplemented by a vacuum chamber above the valve. This both assists the spring and balances the valve against the pump pressure, which tends to hold it open. By this means, and by reducing the weight of the valve, which is of high tensile steel, the Siegener Company have succeeded in producing a valve gear which, as the author can testify, operates quite silently and without appreciable wear or shock.

The piston is a plain iron casting of very simple design, and is, of course, water-cooled, the supply of cooling-water being fed along the piston-rod in the usual manner.

A cross-section of an English Korting piston and rod is shown in fig. 86, and the author is not aware of any important difference between the English and German design of this part. Its great length, coupled with the large amount of water it contains, renders it very heavy.

The crankshaft is of the built-up type, with solid forged mild-steel webs shrunk on to the shaft. The connecting-rod is of the usual marine pattern, except that it has a somewhat peculiar big end. This is not split in the ordinary way at right angles to the rod, but in a plane parallel to the axis of the rod and considerably above the centre line. The design dispenses with the use of long bolts, and is very strong and accessible, but must be considerably more expensive to manufacture than the usual type. The bedplate consists of two main girders cast in sections and held together by means of wrought-iron shrink rings. The cylinder rests on these beds with the plane of the feet on the centre line.

The scavenge-pumps are bolted to the outside of one of the main girders, and both air- and gas-pumps are provided with piston valves, operated from an eccentric on the crankshaft. The valves of the air-pump are set to give a full delivery of air at all times; but those of the gas-pump can be revolved about their rods, and are provided with diagonal slots both in the valve and liner. Thus by rotating them through a small angle the cut-off can be varied, and a greater or lesser proportion of the gas returned to the suction. Owing to the great size and weight of these valves, they are not controlled directly from the governor, but through the medium of a hydraulic relay. As in the case of almost all large engines, starting is effected by means of compressed air, the air being admitted through a small spring-loaded poppet valve fitted in each cylinder cover.

A considerable number of these large single-cylinder engines have been constructed by the Siegener Maschinenbau, and are in successful operation in various parts of Germany. They are particularly suitable for driving blast-furnace blowing-tubs, owing to the wide range of speed of which they are capable.

The English Korting engine, as designed and built by Messrs. Mather & Platt, differs from the German design in its mechanical features, which have been greatly simplified. The use of a side shaft for operating the valves has been entirely dispensed with, and the valves operated direct from an eccentric on the crankshaft, a thoroughly sound mechanical job. The gas- and air-pumps are entirely different from the German design, and their arrangement is very well shown in the diagrammatic section (fig. 87).

It will be seen that there are two single-acting air-pumps and one double-acting gas-pump in between, so arranged as to form one long continuous barrel. Piston valves have been dispensed

with, and replaced by a large number of small flat-plate valves, the suction valves being mounted on the pistons and the delivery valves on the cylinder covers. The arrangement as shown in this diagrammatic section is not altogether above criticism, for with such

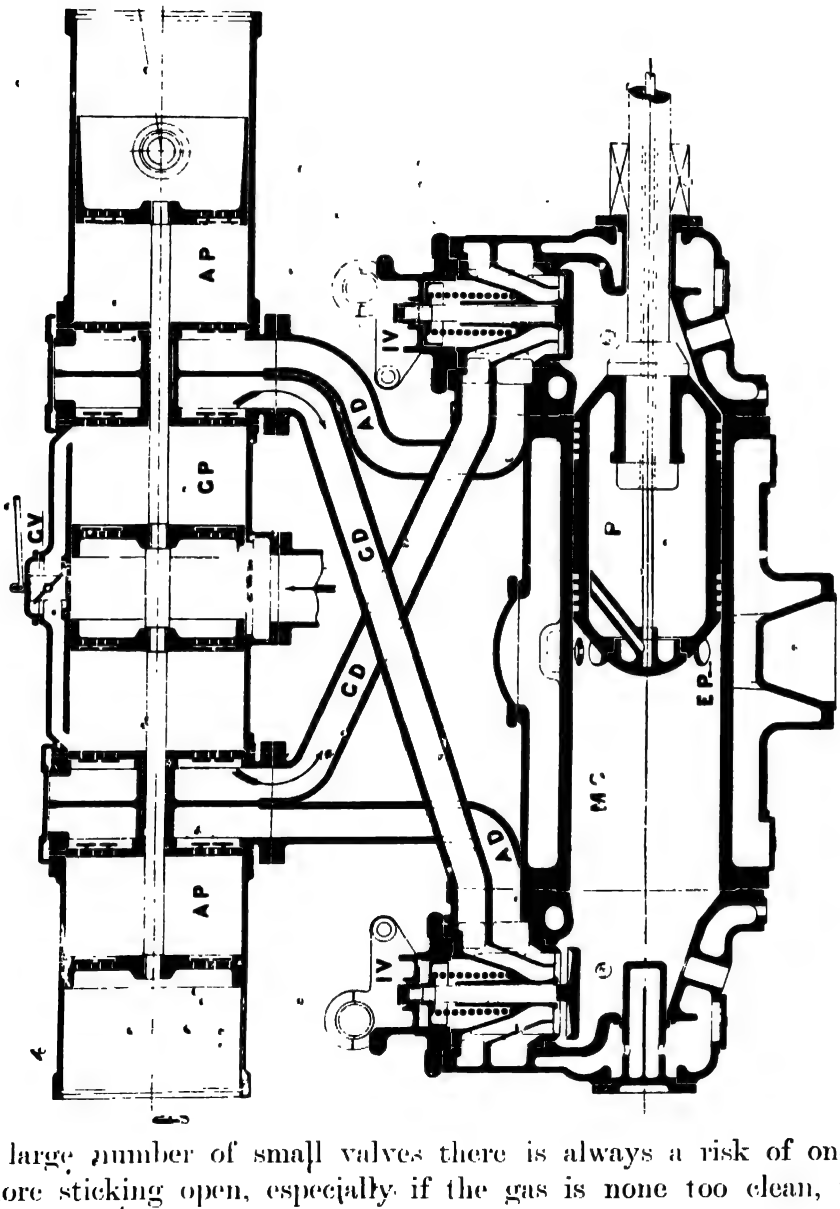


Fig. 87.—Diagrammatic Section of Double-acting Two-cycle Gas-engine

a large number of small valves there is always a risk of one or more sticking open, especially if the gas is none too clean, thus causing unequal distribution and serious loss of power. Apart from this risk, the leakage is bound to be considerable, and access to the pump valves involves dismantling a considerable amount of gear.

In the more recent designs these valves have been replaced by a smaller number of large valves, mounted in cages which can be removed bodily without disturbing any other part of the mechanism. On the inlet side of the gas-pump it will be noticed that a very wide port is provided, the effect of which is that the delivery stroke of the gas piston does not commence until after this port has been covered. The width of the port determines the "lead" on the air and also the maximum quantity of gas, and its effect is precisely the same as in the German design. It can be varied, within certain limits, to suit gases of different heating value, though, of course, by doing so the air lead is also affected. The quantity of gas is further controlled by means of a butterfly valve, which, when open, allows the gas to pass freely from one side of the piston to the other—a system which the author does not greatly favour. It is evident that with a given position of the valve, as the engine speed increases, the proportionate quantity of gas that can pass across the valve is reduced. Hence, there is a tendency for a larger charge to enter the power cylinder with increase of speed, a somewhat unstable condition. Again, if the valve is large, its action is unduly sensitive, and is liable to cause hunting; if small, there is always the danger of its being unable to pass enough gas at high speeds even when fully open. Were this the case, it is evident that the governor would lose control of the engine, and there would be nothing to prevent it from "running away".

If the exhaust back-pressure increased uniformly with the speed, this objection would not apply, for, as the back-pressure rose, the proportion passed by the bye-pass valve would increase and the balance be restored. In practice, however, this cannot be relied upon, owing to the pulsations set up in the exhaust pipe; for, should the periodicity of the exhaust synchronize with the pulsations in the pipe or some function of them, it is possible that, instead of a back-pressure, even a partial vacuum may be formed in the cylinder during the charging period, and an excessive charge of gas drawn in. Again, even at a uniform speed the exhaust back-pressure varies very considerably, due to this pulsating effect. In cases where a very short exhaust pipe can be used, discharging into a large receiver, or exhaust-heated boiler, and where the governing is very close, this system is perfectly satisfactory, but, as a general rule, the author regards it as too sensitive to disturbances in the exhaust. In this respect the method employed

in the Siegener engine, and referred to previously, seems preferable, for the charge of gas is positively measured by the governor, and is not liable to serious variations, even with a widely varying exhaust back-pressure.

By providing very large valve areas in the pumps, and unusually large inlet valves in the power cylinder, also by carefully adjusting the size of the exhaust ports, the pump losses from these engines have been reduced to from 5 to 6 per cent at full load and normal speed. A set of diagrams, illustrated in fig. 88, taken from a single-cylinder engine of 600 horse-power, running at 45 R.P.M., show a pump loss of only 1·9 per cent, a remarkably low figure. Thanks to the great reduction in fluid losses, a mechanical efficiency of from 82 to 84 per cent is obtained — a figure but little below that of a good modern four-cycle engine.

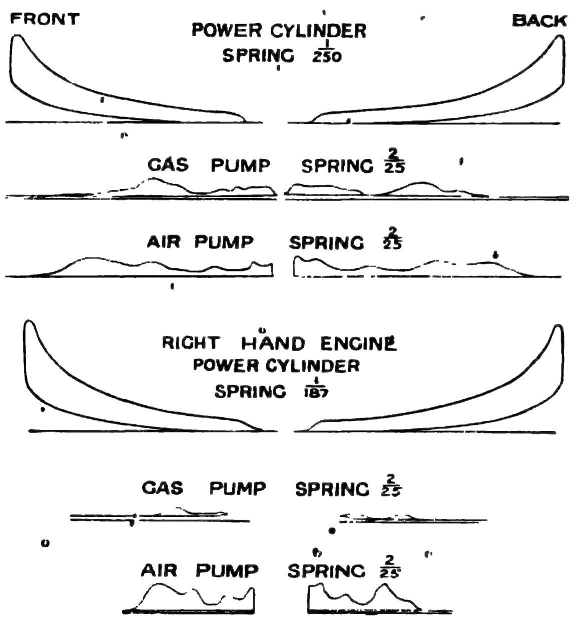


Fig. 88

The cylinder, for all smaller sizes, is usually cast in one piece with the water-jacket, no separate liner being employed, but in order to ensure a hard wearing surface for the piston and rings the inside of the cylinder bore is "chilled". For larger sizes, two separate liners are employed, as in the Siegener cylinder. The cylinder covers are similar in most respects to those of the Siegener engine, and are made interchangeable, a blank water-cooled plug being inserted in the back cover to fill the opening for the piston-rod. No tail-rod is employed except when the engine is used for driving a blowing-tub or pump. The use of a tail-rod is considered to be unnecessary owing to the great length and large bearing surface of the piston, the under side of which is padded with white metal. To prevent risk of cutting this metal, and also of scraping off the lubricant, no exhaust ports are

provided in the under side of the cylinder. The crankshaft is built up, the webs and balance weights being of cast steel shrunk on to the shaft and crank-pin; no keys or steady pins are fitted, shrinkage alone being relied upon. This is perfectly satisfactory provided that the crank-webs are deep enough to withstand the bursting stress due to the large degree of shrinkage necessary.

The remainder of the engine calls for very little comment, since it is built on the most approved steam-engine lines; great strength, simplicity, and low cost of manufacture being aimed at throughout. Compared with the German, the English Korting is certainly a very much neater and more workmanlike job, and is undoubtedly superior in mechanical efficiency.

The Duplex Engine.—The Duplex engine, which has been recently placed on the market by Messrs. Mather & Platt, is built to the designs evolved by Mr. Allan Chorlton, and, in the author's opinion, represents a very important advance. Like the Fullagar it is both simple and efficient, and above all it seems to offer a possible solution to the problem of the really high-powered gas-engine, for it is double-acting, and at the same time it is capable of operating with a high mean pressure and at a high piston speed. Thus the specific power of each cylinder is very great, while, owing to the special design of the cylinders, little trouble need be feared from unequal expansion, even with very large cylinders.

Fig. 89 shows a vertical section of a twin-unit Duplex engine, designed to develop 1000 B.H.P. at a speed of 300 R.P.M. It will be observed that each unit consists of two double-acting power cylinders, connected together top and bottom by common combustion chambers. Around the middle of each cylinder is a belt of ports, as in the Korting engine; one set of ports is for the exhaust, and the other for the gas and air inlet. In these two cylinders are pistons, which reciprocate together, one controlling the inlet and the other the exhaust ports; the piston controlling the latter has a lead of about 15 degrees ahead of that controlling the inlet. The exhaust ports, moreover, are slightly deeper, so that, owing to their lead and their greater depth, they are opened some 19 to 20 degrees ahead of the inlet ports, and are closed some 10 to 12 degrees before them.

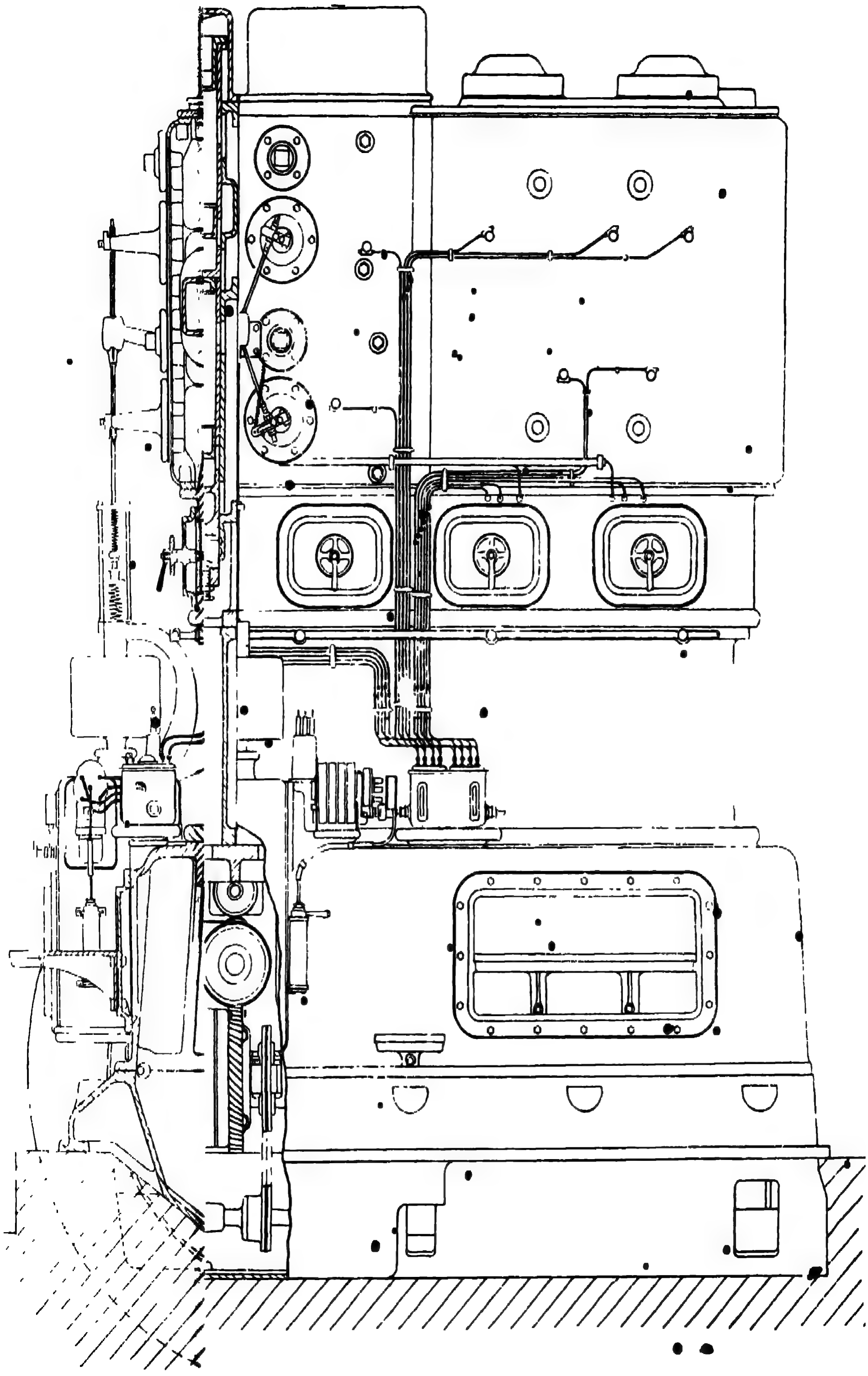
Gas and air are supplied to the inlet ports by means of double-acting reciprocating pumps of peculiar form, for, in addition to the pistons, the inner wall of the cylinder also reciprocates, and in doing so opens or closes ports cut in the outer cylinder wall. This form of

valve gear, known as a sleeve valve, has been successfully employed in the light high-speed petrol-engines used for motor-cars, and has much to recommend it for this purpose, provided that the gas is clean and reasonably free from tar. It offers a very large port area, and is positive in its action. Moreover, it is possible with this arrangement, without any increase of complication, to give a secondary air-scavenge for the purpose of clearing the ports and passages of any gas that may be lingering there. As in the Korting engine, care is taken to ensure that the gas and air shall not come into contact with one another outside the cylinder, and not only are separate passages employed, but a separating rib is fitted round the outside of the cylinder surrounding the inlet ports.

One of the most important features of the design is that the cross-section of the combustion chamber at any point is circular, and consequently the thickness of metal can be reduced to a minimum, hence the internal stresses due to the temperature difference between the inner and outer surfaces are also reduced to a minimum. These stresses are still further reduced by the absence of any large bosses, valve pockets, passages, or flanges which necessitate a change of thickness at any one point. Since it is these stresses, due to temperature difference and pressure, that in practice limit the size of cylinder, it is reasonable to suppose that very much larger cylinders could be successfully employed in this type of engine than is possible with other types. At the same time, owing to the fact that the cylinder is double-acting, a very high specific power can be obtained from a comparatively small cylinder.

Reference to the sectional drawing will show that the cylinder is constructed in three parts, the central part consisting of the exhaust and inlet belts, into which the halves of the cylinders are spigoted. In this design the only flanges required for the cylinders are situated close to the ports, and therefore at the point of lowest temperature, a construction which permits of free expansion in any direction. All risk of cracking the water-jacket, due to the greater expansion of the inner shell of the cylinders, is completely avoided by casting the cylinders without any water-jacket whatever, and simply immersing them bodily in a large tank. For convenience of manufacture, the two halves of the cylinder are identical, the openings for the piston-rods in the top half being filled by means of hollow plugs, which can be water-jacketed if necessary.

The pistons are machined from mild steel and are made in two halves, bolted together round the circumference. Besides the great



reduction in weight that can be effected, mild-steel pistons have the advantage of much greater reliability than cast-iron, for not only can they be made thinner, and hence the temperature difference between the inner and outer walls, and consequent stresses, reduced, but the material itself is better able to resist such stresses. Their use, however, for large engines is only possible when, as in this case, they are relieved from all bearing pressure by means of external cross-heads.

The governing of the engine is effected by throttling the supply of both gas and air simultaneously; governing the gas alone was found to be insufficient, owing to the great amount of diffusion that takes place in the inlet cylinder; only by controlling both the gas and air proportionately has it been found possible to obtain regular running on the lightest loads. It has already been pointed out that it is desirable to reduce the supply of air on light loads in any case, for by doing so the fluid losses can be reduced and the mechanical efficiency improved.

The crankshaft of this engine is necessarily somewhat peculiar. No bearing is provided between the two cylinders in the latest design, in order to keep them close together and reduce the length of the combustion chamber as far as possible. Owing to the lead of the exhaust piston, the crank-pins for the two pistons are out of line to the extent of about 15 degrees. Such a shaft would be most difficult and costly to forge and machine from mild steel, but the difficulty has been overcome by employing a single steel casting for the two crank-webs, balance-weights, and crank-pins, this casting being bored out and shrunk on to the sections of the main shaft. The employment of a steel casting in this manner is perhaps a somewhat daring departure, but it appears to have been justified.

Apart from the crankshaft, the rest of the engine conforms almost exactly to modern high-speed steam-engine practice, an enclosed crankcase being employed, with forced lubrication to all bearings. In fact, as an illustration of the great similarity between this engine and an ordinary vertical steam-engine, it is worthy of note that the first Duplex engines built were converted from ordinary triple-expansion steam-engines, and that the only serious alteration necessary below the entablature was the provision of a new crankshaft. The high-pressure cylinders were replaced by the scavenging-pumps, and the intermediate and low pressure by the two cylinders comprising the Duplex unit. The same cross-heads,

slides, connecting-rods, bedplate, and standards were employed without any alteration. These original engines, which have a maximum power of about 700 B.H.P. each when consuming Mond producer gas, are at work at the Castner Kellner Alkali Works at Runcorn.

The principal objection to the Duplex engine, in the author's opinion, lies in the want of balance. Since both pistons are virtually

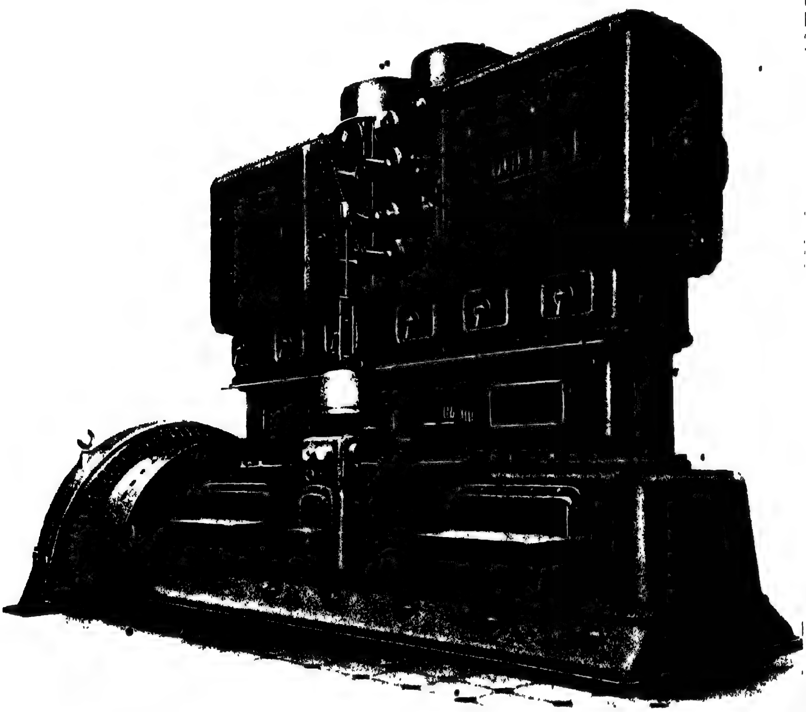


Fig. 90.—1000-B.H.P. Duplex Patent Valveless Gas-engine

connected to the same crank-pin, and rise and fall together, the balance is no better than a single cylinder whose reciprocating parts are equal in weight to the two sets employed in the Duplex. This objection can be mitigated to some extent by the employment of two units with cranks at 180 degrees; but, even so, there is a large unbalanced couple, due to the distance apart of the centre lines of the two units. Fig. 90 shows a photograph of the 1000 horse-power Duplex gas-engine; its neat and compact appearance is very

striking, but it will be observed that the distance between the two units is very great, owing to the fact that the scavenging cylinders have been placed between and not outside them. No data are available as to the mechanical efficiency, thermal efficiency, or pump losses of these engines. It is by no means an easy matter to obtain an accurate indicator diagram from engines using two pistons slightly out of phase, for it is necessary to drive the indicator from some gear whose motion is at all times an exact mean between that of the two pistons. Such a gear is too costly to fit on an ordinary engine built for commercial purposes, and consequently, until recently at all events, no accurate diagrams have been taken from a Duplex engine.

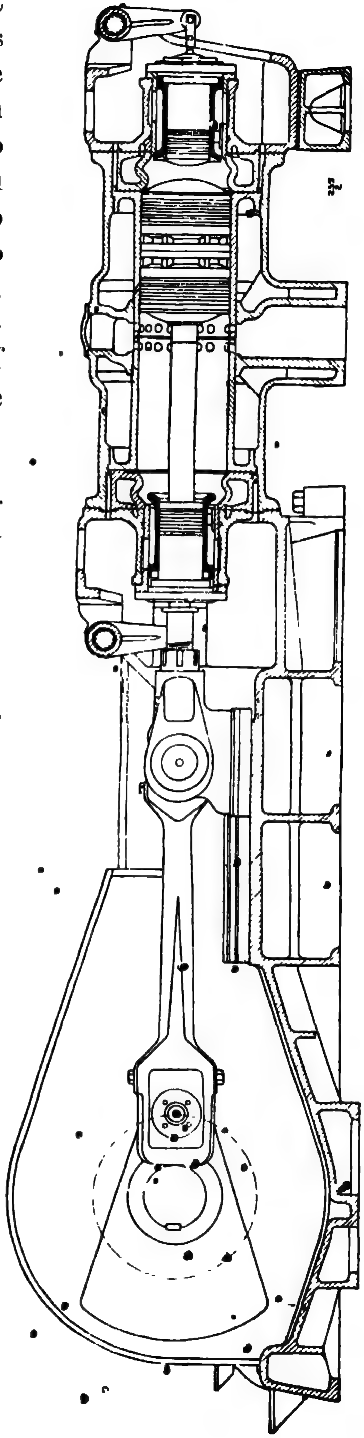


Fig. 91.—Sectional Elevation of Cylinders and Bedplate of Illmer Gas-engine

Illmer Gas-engine.—The Illmer gas-engine, built by the Reading Iron Works of Pennsylvania, U.S.A., is an entirely new machine designed somewhat on the lines of the Korting engine, but containing certain radical departures from that design which make it most interesting. Fig. 91 shows a sectional elevation, fig. 92 a diagrammatic section illustrating the action of the combined air-scavenging and charging pump, and fig. 93 a section of the water-cooled piston.

Referring first to the sectional elevation, it will be observed that the inlet valves are mounted horizontally in the ends of the cylinders and are supported by long internal sleeves, which, owing to their ample bearing surface, remove the common objection to large horizontal valves.

The valve-carrying sleeve at the front end of the cylinder serves also as the stuffing-gland for the piston-rod. These valves, it will be observed, have no springs, and are opened and closed mechani-

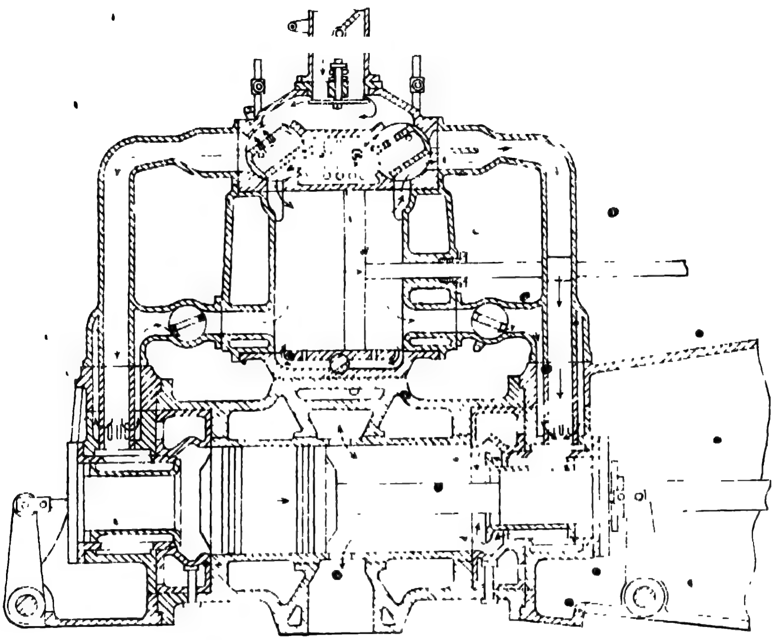


Fig. 92.—Diagram of Scavenging and Charging Pump of Illmer Gas-engine

cally by means of a toggle action which must require very careful adjustment. The combustion chamber is in the form of a plain water-jacketed distance piece, and presents very little surface to the burning gases, so that the heat loss to this part should be comparatively small. On the other hand, the shape is hardly conducive to good scavenging, and, in the author's opinion, there will be a tendency for the incoming gas to sweep along the walls of the cylinder, and so out through the exhaust ports, leaving the central core of exhaust gases undisturbed. The position of the igniter, always an important consideration in two-cycle explosion engines, is not shown.

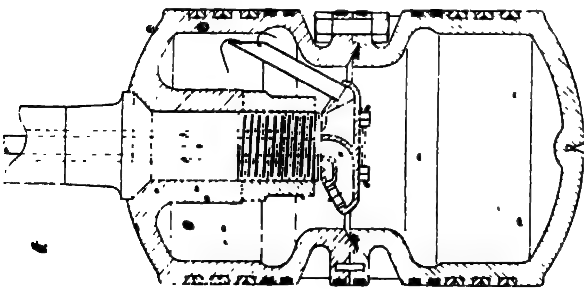


Fig. 93.—Section of Water-cooled Piston of Illmer Gas-engine

From a mechanical point of view the design of the combustion chamber and cylinder cover seems excellent. Both are simple castings, which should be free from initial internal stresses, and, being circular in cross-section, may be made of uniform thickness and comparatively thin. The cylinder covers proper are not exposed to the heat of combustion, for they are shielded by the valves and the valve-carrying sleeves, the latter being plain water-cooled cylinders. The cylinder body itself follows the usual Korting design, with two separate liners pressed in from either end, and almost butting in the centre.

The scavenge-pump is ingenious. A single-pump cylinder is used both for the primary air-scavenging and for introducing the mixture. This is accomplished by filling the passages connecting the scavenging-pump and the main inlet valves with pure air at the end of each scavenge stroke, which air is forced into the cylinder, ahead of the combustible mixture, at the commencement of the next stroke. Looking at the diagrammatic section, it will be observed that the passages leading to the main inlet valves contain an internal pipe for part of their length; this internal pipe is connected to the pump cylinder through an oscillating valve, which puts it alternately in communication with the pump cylinder and the gas supply. The external annular portion of the passage is in direct communication with the pump at all times, and is also in communication with the inner pipe, by means of a number of holes drilled round the latter, near the end where it joins the main inlet valve. When the pump piston travels out on the suction stroke, the oscillating valve on the inner pipe is open to air, which is drawn in through this pipe, thence through the holes in the outer pipe, and so into the pump cylinder, filling the whole passage with air. At a certain point in the piston stroke the oscillating valve changes over, cutting off communication between the inner pipe and the air, and opening a direct port between the pump cylinder and the mixture supply. Thereafter, till the end of the stroke, combustible mixture is drawn into the pump.

On the delivery stroke, the oscillating valve closes the inner pipe both to gas and air, and puts it in communication with the pump. As soon as the main inlet valve opens, the contents of the passages, which consist of pure air, are first driven into the cylinder, followed by the charge of combustible mixture. Near the end of the stroke, the piston overruns a small port, which allows the combustible mixture entrapped in the clearance space to pass round to

the other side of the piston, thus avoiding any idle stroke due to the gases re-expanding in the clearance space. The whole scheme is exceedingly ingenious, but it is doubtful whether it is really worth doing, for, although mechanically fairly simple, it involves passing the air and gas through somewhat tortuous passages at a high velocity and with violent reversals of direction. This must cause a very considerable amount of fluid friction, due to two separate pumps.

The idea of making one single piece of mechanism perform several different functions is always attractive, and especially fascinating to anyone of an inventive turn of mind, but as a general rule such combinations do not turn out to be practically or commercially sound. Such a combination scavenging-pump, as this is bound to be very sensitive to sudden changes in the exhaust back-pressure, and, in the author's opinion, it is very doubtful whether it will give such good results as either a single pump delivering combustible mixture without any air-scavenge, or separate air- and gas-pumps.

Governing in this engine is carried out by means of a bye-pass valve on the pump cylinder, in much the same manner as in the English Korting.

In common with the English Korting, the valve gear of the Illmer engine is driven direct from the crankshaft by means of an eccentric, and is evidently modelled upon the standard American Corliss steam-engine gear, which it resembles very closely. The piston of the Illmer engine resembles that of the Duplex, and is constructed in two halves bolted together round the circumference, but unlike the Duplex it is made of cast iron. The bearing surface appears to be somewhat inadequate for a horizontal engine in which no tail-rod is fitted, and in which the whole of the weight of the piston has to be carried by the liner. The system of making the piston in two halves, bolted together in the middle, though excellent for vertical engines, is not very suited to the horizontal type owing to the great reduction in the bearing surface which it involves. It will be noted that, as illustrated, the bolts in the piston could not be inserted unless separate nuts were fitted at each end, always a troublesome arrangement.

Apart from the cylinder and scavenge pumps, the engine follows the usual conventional design for American steam-engines. An overhung crank is employed, which is standard practice in America for almost all horizontal steam- and gas-engines. It certainly reduces the cost of the crankshaft, and permits of the use of a solid box-

type bearing for the big-end of the connecting-rod. On the other hand, it necessitates a very much heavier bedplate, since the stresses are not carried directly from the cylinder to the crankshaft.

The following tests have been carried out upon a 300 horse-power Illmer engine and producer:—

Duration of test	$33\frac{1}{2}$ hours.
Grade of coal	Westmorland bituminous, at 14,100 B.T.U.s per pound.
Average load	284 lb.
Coal consumption (per B.H.P. hour) ...	1·14 lb
Producer efficiency on lower heat value of gas	64 per cent.
Combined thermal efficiency of engine and producer	15·7 „
Brake thermal efficiency of engine ...	24·8 „

Indicator cards taken during these tests are shown in fig. 94. The brake thermal efficiency of the engine is distinctly low for a

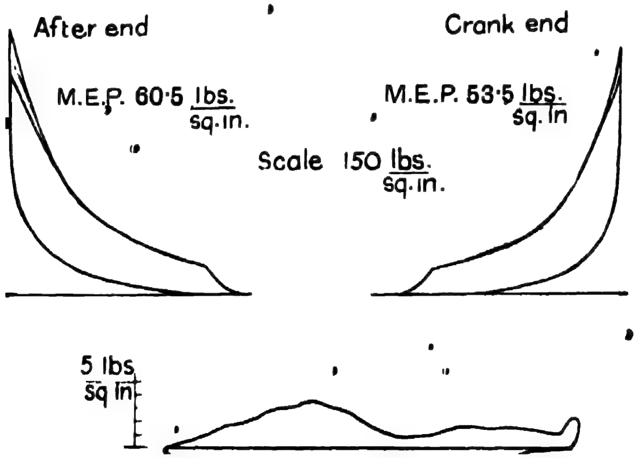


Fig. 94. Illmer Gas-engine. Diagrams from working Cylinder and Pump

machine of this size, and this is probably due in a large measure to the fluid losses in the pump cylinder, and perhaps to incomplete scavenging due to the unsuitable shape of the combustion chamber.

CHAPTER XIX

SMALL TWO-CYCLE ENGINES

The engines already described may be taken as fairly representative of the leading types of large two-cycle gas-engines; there are, of course, numerous other examples on the market, but they differ from the foregoing only in details.

It will now be well to examine some of the smaller types of two-cycle constant-volume or explosion engines, such as are used for motor boats, automobiles, &c. These engines run at very high rotative speeds, and consequently entirely new factors are introduced into the problem. The difficulty of obtaining a good mechanical efficiency is very much intensified in these small engines, owing to the inertia of the gases in the pipes and passages, which increases the fluid losses enormously. Good rotary balance becomes, in the author's opinion, a matter of paramount importance, for not only is such balance essential from the point of view of mechanical efficiency at high speeds, but it must also be remembered that these engines are generally very light in themselves, and are often mounted on a very light structure, such as a boat or automobile. In such cases any vibration will cause serious injury both to the engine itself and the rest of the machine, besides being most objectionable to the occupants. Also, owing to the high speed, the increased inertia effects of the reciprocating parts throw severe loads on the bearings, in some cases considerably in excess of the fluid pressure, causing excessive friction and wear unless special attention is paid to the lubrication of these parts. In one respect, however, the two-cycle engine is better off than the four-cycle, namely that, owing to the greater number of impulses, the fluid pressures bear a greater ratio to the inertia pressures.

In small high-speed engines, on the other hand, the question of exposed surface in the combustion chamber becomes of little importance, as also does that of internal stresses in the metal walls, for these are, in any case, so thin that the temperature differences

between the inner and outer surfaces are very small. With such engines the questions of primary importance are therefore (1) good rotary balance, (2) light reciprocating parts, (3) free flow for the gases, without sudden bends or violent reversals of direction, (4) a careful study of the pulsations or pressure oscillations in the exhaust pipes and other passages.

A number of small petrol-engines on the principle of the Duplex engine, suitable for automobile and marine purposes, have recently been introduced, employing the inverted **U**-type of cylinder, such as the Lucas and Lamplough. These engines differ in certain respects according to the aims of the designers. Mr. Lucas and Mr. Lamplough have considered that rotary balance is the fundamental consideration, and have designed their engines with a view to securing such balance, while Mr. Lucas has gone still further, and, with the aid of two crankshafts, has secured reactionary balance as well.

The Lucas Engine.—The Lucas engine is illustrated in figs. 95 and 96, from which it will be seen that the two pistons are each connected to a separate crankshaft, the two shafts being geared together by means of teeth cut round the outside of the crank-web. Each shaft is mounted with a fly-wheel of equal weight and equal moment of inertia, but the drive is taken from one shaft only. By this means perfect reactionary balance can be obtained, for the reaction due to the power stroke, instead of tending to rotate the whole engine around the crankshaft, tends to force the two cylinders apart, but produces no displacement of the whole mass of the engine. By the use of two crankshafts, also, a very fair rotational balance can be effected, if balance weights equal in mass to the reciprocating parts, and at the same radius, are attached to each crank; but the error due to the angularity of the connecting-rods still remains. In fact, the primary forces are balanced, but not the secondary, the condition being the same as in the two-cylinder Fullagar engine.

Dealing first with the cycle of operations, it will be observed that, unlike the Duplex engine, the exhaust piston has no lead, and, consequently, very deep exhaust ports have to be used. These ports actually open no less than 66½ degrees before the bottom of the stroke, which is equivalent to a loss of stroke of nearly 30 per cent. The inlet ports are uncovered only 11 degrees later, which represents a very small amount of lead when expressed in degrees of the crank; but it must be remembered that with deep exhaust ports the opening is very rapid, owing to the high velocity of the

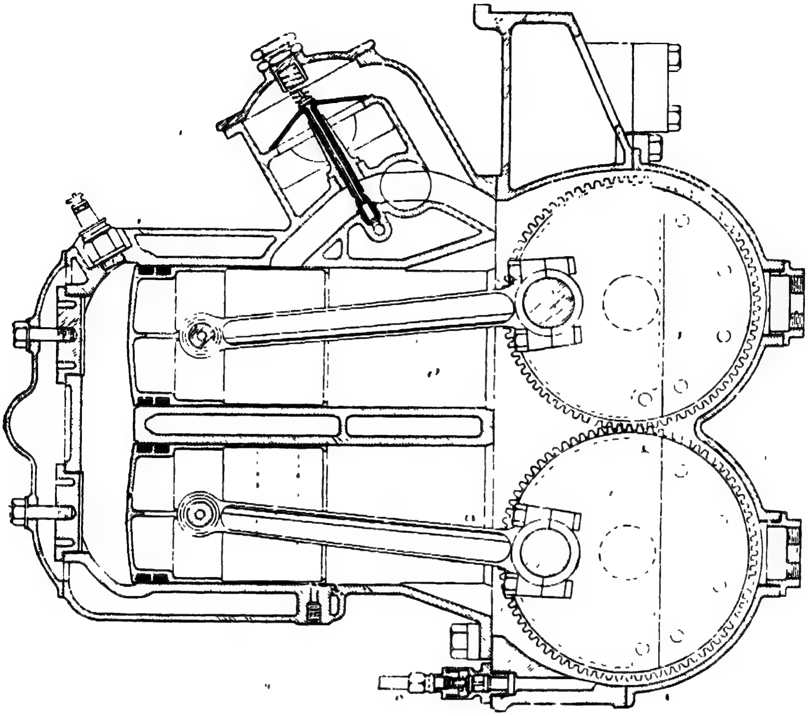


Fig. 96

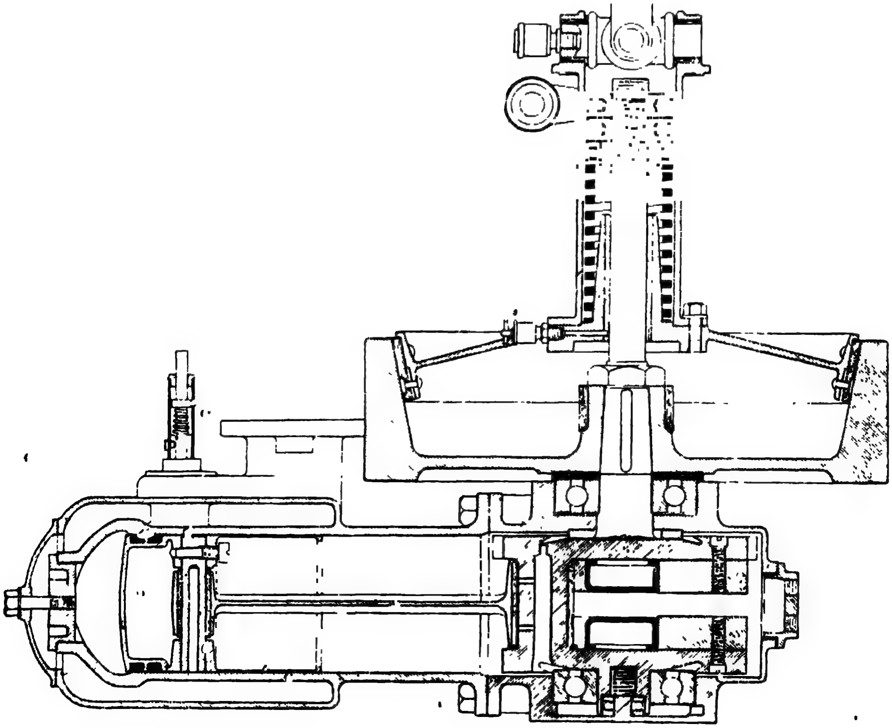


Fig. 95

15-H.P. Lucas Valveless Engine

piston at this period of its stroke. Expressed in percentage of the stroke this lead amounts to about 8 per cent. More efficient scavenging could probably be obtained if the design of the combustion chamber were modified to the form of a gradually expanding cone. For scavenging, crank-chamber displacement is employed, a system which has already been discussed. In this particular case the clearance space has been reduced to a minimum by fitting, in addition to the balance weights, hollow aluminium blocks to fill up as much of the space between the crank disks as possible.

On the upward stroke of the pistons, air enters through the large spring-loaded mushroom valve, and passes into the crank-case. At the same time a thin tapered needle, which is attached to the valve, and which normally closes the petrol jet, is raised, and petrol is allowed to flow into the connecting passage between the inlet ports and the crankcase, where it vaporizes. As the pistons descend, the mushroom valve and the needle valve are closed, and the air within the crankcase is compressed until the inlet ports are uncovered, when it passes into the cylinder together with the petrol vapour. It is obvious that by this means of scavenging, unless there is complete diffusion between the air and petrol vapour, the latter will be driven into the cylinder ahead of the air, and any gas that is lost through the exhaust ports, will be a rich mixture of petrol vapour and air. With crank-chamber scavenging, however, the scavenging efficiency is poor, and the volume of working fluid entering the cylinder small. It is probable, therefore, that very little fuel reaches the exhaust ports, and that the loss from this source is trifling.

The primary object in separating the air and petrol vapour during the suction stroke is to prevent the charge in the crank-chamber from being ignited when the inlet ports are first uncovered. This is a risk which is always liable to occur when, from defective carburation, or other causes, combustion in the cylinder is delayed, and the temperature of the gases in the cylinder, at the time when the inlet port is opened, is above the ignition temperature of the fresh charge. Mr. Lucas has sought to prevent this occurrence by interposing a layer of petrol vapour in the passage, which does not contain sufficient air for combustion.

Control of the engine is effected by means of the throttle valve in the connecting passage— a very good arrangement, in that the velocity of the gases through the inlet port is reduced proportionately with the load, and hence the degree of diffusion is also

lessened. Moreover, with the throttle partially closed, the mixture immediately over the inlet piston will be richer and more readily ignited.

Considering the mechanical features, it will be observed that the crankshaft is mounted on ball-bearings — a very excellent arrangement for an engine of this description, where the lubrication of the main bearings presents considerable difficulty. Ball-bearings, if kept free from rust, will run for a great many years without any measurable wear, and with the most meagre lubrication. They are open to the objection, however, that they cannot be made really silent, and all engines fitted with ball-bearing crankshafts are distinguishable by a slight "growling", which is exceedingly difficult to locate. In this case, however, the noise of the ball-bearings will probably be indistinguishable from that due to the crankshaft gears, although the author feels bound to admit that these gears are surprisingly quiet in practice, at all events when the engine is new and in good condition. Leakage of air through the bearings is prevented by means of faced washers between the crank-web and the wall of the crankcase. These washers are kept bearing hard against the walls of the crank-chamber by the simple device of using single helical gears for the crank disks, and, taking advantage of the unbalanced thrust from these gears for this purpose.

For the lubrication of the connecting-rod bearings, concentric grooves are turned in the outside of the crank disks, into which a small amount of oil is fed. From these grooves it is driven by centrifugal force through the hollow crank-pin to the big-end bearings, and from thence it is conducted to the small-end bearings by means of light copper pipes. It will be noted that the cylinders are not mounted directly over the crankshaft, but that their centres are considerably closer. This is done for two reasons: (1) To reduce the length of the combustion space as far as possible, and (2) to reduce the angular thrust of the connecting-rod upon the cylinder wall during the expansion stroke, a somewhat doubtful advantage. The greatest care has been taken to reduce the weight of the pistons and connecting-rods to the lowest possible limit, and with this end in view the gudgeon-pin is fitted considerably higher up the piston than is customary. The most striking feature about the Lucas engine is its extreme simplicity and neat and compact appearance. The two cylinders, together with their water-jackets, connecting passages, and carburettor, are all cast in one piece, while

the crank-chamber is of the simplest possible design. In practice the engine runs well over a good range of speed, but owing to the use of crank-chamber displacement the mean pressure is very low, and falls off rapidly with increase of speed, while very high speeds are unattainable from the same cause. The fuel consumption is low for an engine of this type. The engine is fairly silent, and the balance excellent except at the highest speeds, when the want of secondary balance is observable; but owing to the extreme lightness of the reciprocating parts this is comparatively slight. The absence of any rotary recoil is most noticeable, especially at low speeds, but the fitting of an extra fly-wheel adds considerably to the weight of the engine, which is already above the average for engines of the motor-car type.

The maximum power obtained from this engine is approximately 20 B.H.P. The cylinders are $5\frac{1}{4}$-in. bore and $5\frac{1}{2}$-in. stroke, but, owing to the low brake mean pressure, which ranges from 58 lb. per square inch at 200 R.P.M. to 20 lb. per square inch at 1500 R.P.M., the specific power in relation to the size of the cylinder is low when compared with a four-cycle engine of equal cylinder dimensions, and amounts to only 1 B.H.P. for every 11 cu. in. of swept volume. The Lucas engine in its present form was first placed on the market in 1905, and since that date the design has not been changed to any appreciable extent.

The Lamplough Engine.—The Lamplough engine, illustrated in figs. 97 and 98, is a recent production, and contains many original and ingenious features. Its designer has evidently considered that rotary balance is of the first importance, and, with this end in view, he has sought to balance the power pistons by means of a composite pump piston of equal weight, connected to a second crank at 180 degrees to that of the power crank. The arrangement necessitates the use of a receiver between the pump and working cylinders, always a bad feature in two-cycle engines which scavenge with combustible mixture; for, if the receiver be made of very small capacity, the pump losses become excessively high, owing to the high pressure generated in it, and released when the inlet ports are opened. This high pressure, besides incurring excessive pump loss, produces a high velocity through the inlet ports, causing an unnecessary amount of diffusion between the fresh charge and the products of combustion. If, on the other hand, the receiver be made of large capacity, excessive condensation and precipitation will take place in it, and a fire-back into it will be serious, for it will foul the fresh

charge from the pump for a considerable number of revolutions, and not only cause the engine to miss fire, but produce such slow burning of the next few charges as to incur the risk of a second fire-back through the inlet ports, and so on until the engine is either stopped or its speed very much reduced. Of the two evils the use of a small receiver is the less objectionable, and in the Lamplough engine the receiver is of extremely small capacity, hence the fluid losses are high.

Turning to the sectional drawing, it will be observed that the

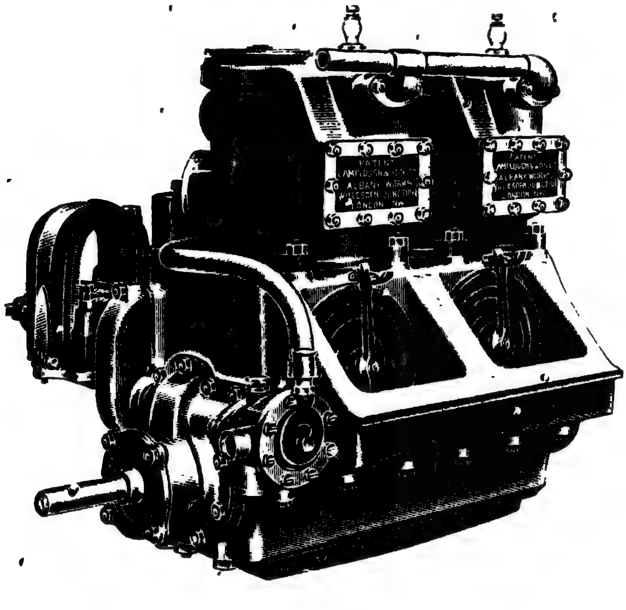


Fig. 97. –Lamplough Engine

two cylinders are placed across the crankshaft, and the two pistons coupled to the same crank-pin by means of bent connecting-rods. This has the effect of giving a lead to the exhaust piston without necessitating the use of a two-throw crank, but the angularity of the connecting-rods is necessarily very great. The design of the combustion chamber is similar to that of the Lucas engine. As in the Lucas engine, the pistons are pressed from mild steel, and are exceedingly light. The pump cylinder is placed alongside the power cylinders, and has a curious and ingenious composite piston connected to the crank-pin by means of two connecting-rods.

This piston consists of two parts: (1) an outer sleeve, to which one of the connecting-rods is attached, and which bears on the cylinder walls only at its two ends, the middle part being recessed,

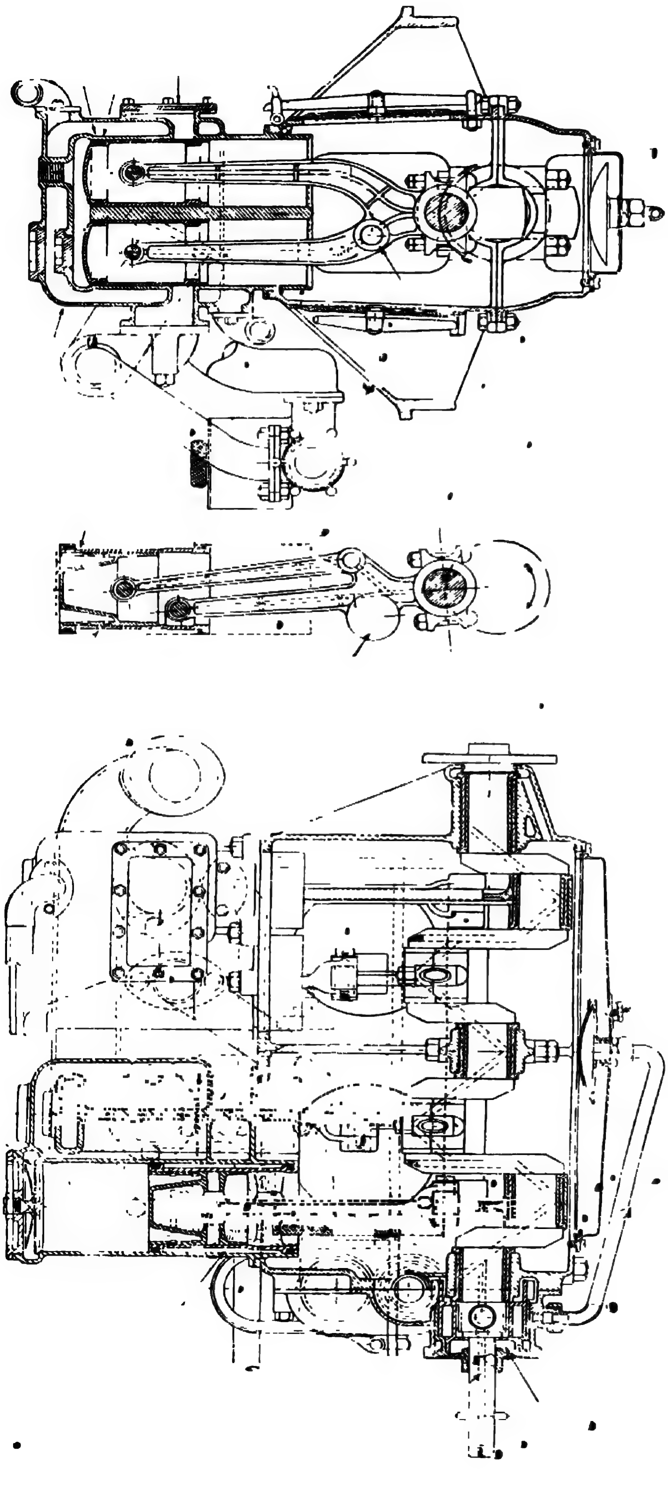


Fig. 98.—Diagrammatic Sections of Lamplough Engine

in order to leave an annular space between the sleeve and the cylinder; and (2) an inner piston attached to the other connecting-rod. The two portions of this piston reciprocate together, but owing to the different angularity of the two connecting-rods, there is a small relative motion between them, and this motion is taken advantage of to admit the charge from the carburettor to the cylinder. Ports are cut both in the sleeve and in the wall of the piston, in such a position that the two sets register during the downward stroke, but, owing to the slight relative movement, are masked during the upward or delivery stroke. The induction pipe from the carburettor is led direct to the wall of the pump cylinder, in such a position that the gas enters the annular space between the outer piston and the cylinder wall, and is drawn into the cylinder when the two sets of ports register on the downward stroke. This method of combining the inlet valve and piston is simple and ingenious, and the increased weight in this case is no objection, since it is necessary, from the point of view of balance, that the pump piston should be the same weight as the sum of the two working pistons.

The delivery valve of the pump cylinder consists of a disk of exceedingly thin sheet phosphor-bronze, held down to its seating by means of a curved keep-plate, as shown in the illustration; the seating is slotted with a number of rectangular slots. The action of this valve is precisely the same as that of an ordinary leather or rubber pump valve. Such valves are admirably suited to the needs of high-speed two-cycle engines; for, being extremely light, they are capable of very rapid action without noise or appreciable wear, and they cause the minimum of fluid friction. If a suitable bronze be selected and subjected to the proper heat treatment, the life of the valve is very great, and fracture due to fatigue almost unknown.

After passing through the delivery valve, the combustible mixture is directed, by means of passages cored in the cylinder casting, to the receiver—a small chamber surrounding the inlet ports—where it remains until the ports are uncovered. It will thus be seen that the gases, after passing at a high velocity through the pump delivery valve, are brought to rest in the receiver, and again accelerated to a high velocity through the inlet ports. The effect of this is not only to increase the fluid losses considerably, but, when petrol is used as the fuel, the change from high velocity through the pump valve to comparative stagnation in the receiver will result

in the precipitation of a considerable amount of liquid petrol. A portion of this liquid petrol may be re-evaporated, but much of it will enter the cylinder in the liquid form, and, clinging to the cool walls, will never come into contact with the necessary air for combustion. With paraffin as fuel this trouble will be even more accentuated.

With regard to the actual results obtained from a Lamplough engine, the author cannot do better than quote the figures published by Mr. R. W. A. Brewer in a paper read before the Society of Engineers in 1911. Before doing so, however, it is only fair to mention that the tests referred to were carried out on a new engine, and almost, if not quite, the first of its type ever built. For this reason the figures must not be taken as representing the best of which the engine is capable.

The engine tested by Mr. Brewer was a double unit having two pairs of power cylinders and two pumps connected to a four-throw crankshaft with all cranks at 180 degrees, as in an ordinary four-cylinder four-cycle engine.

The bore of the power cylinders was $2\frac{1}{2}$ in. and the stroke $3\frac{1}{2}$ in.; giving a swept volume of 34·3 cu. in. in each unit, or 68·6 cu. in. per revolution of the engine. The clearance space and compression ratio are not stated, but the compression pressure is given as 88 lb. per square inch, from which it may be deduced that the compression ratio was probably about 4·55 : 1. This is based on the assumption that the value for the index is 1·28 for the compression curve, a figure which the author has frequently obtained from small high speed two-cycle engines. The volume of the clearance by this calculation amounts to 9·7 cu. in., and the total volume of each pair of cylinders, when the pistons are at their lowest position, is 44 cu. in.

The bore and stroke of the pump cylinders was $3\frac{3}{8}$ in. by $3\frac{1}{2}$ in., giving a swept volume of 31·5 cu. in.; the ratio between the volume swept by the power pistons and the scavenge piston, is therefore 1 : 0·92, a very low ratio indeed. The volume of air taken into the pump cylinder was measured by means of an anemometer; but Mr. Brewer states in his paper that he is not satisfied with these measurements, and since the volumetric efficiency of the pump cylinder, when running at 1560 R.P.M., is given as 100 per cent, it is hardly likely that they are accurate. With carefully designed pipe-work, however, and ample valve area, very high volumetric efficiencies can be obtained even at such speeds as 1500 R.P.M. But in this case

the author is very doubtful whether as high a volumetric efficiency as 80 per cent is obtainable.

No indicator diagrams were taken during these tests, but the power required to rotate the engine at 900 R.P.M. was tested by driving it from an electric motor. This method will give a fairly accurate idea of the friction losses, but will under-estimate the fluid losses, since, when the engine is not firing, it is probable that the back-pressure on the pumps is considerably relieved; in this respect a two-cycle engine differs from a four-cycle, in which the fluid losses are actually less when the engine is firing. By motoring the friction loss at 900 R.P.M. was found to amount to about 1·70 horse-power, and the fluid losses to only 0·80 horse-power. The former loss will probably be increased by about 10 per cent, and the latter by about 50 per cent, when the engine is running normally, judging from the author's experience. The maximum B.H.P. at 900 R.P.M. was about 10·5. Adding to this 1·9 horse-power for friction, and 1·2 for fluid loss, the indicated horse-power becomes 13·6, and the mechanical efficiency 77 per cent; a very fair result for an engine of such small dimensions, and employing a receiver of small capacity. Of the 23 per cent lost, 14 per cent is accounted for by friction, and 9 per cent by fluid losses.

The table on p. 267 gives the result of the series of tests carried out on this engine.

It will be observed that these figures give such widely varying results, due to the different conditions under which the engine was running, that they cannot be reconciled with each other, and useful information can only be obtained by selecting one test and following it through. By far the best of these tests is C 12, taken when the engine was running at 1370 R.P.M., and developing 15·0 B.H.P. At this speed the mechanical efficiency will probably have dropped to about 72 per cent, for it has been explained in Vol. I that the friction losses increase nearly as the square of the speed, owing to the greater bearing pressures resulting from the increasing inertia of the reciprocating parts, and the fluid losses increase in a somewhat similar ratio, due to the higher velocity of the gases. If 72 per cent be accepted as the mechanical efficiency (the author is taking this figure on the assumption that the ratio of mechanical efficiency to speed will follow the same general curve as in the case of his own tests), then the indicated horse-power becomes 20·8, and the mean effective pressure $\frac{20 \cdot 8}{15} \times 63 \cdot 2$ lb. per square inch = 87·7 lb. per

Test	Revs. per min	Petrol Consumption: Gall per hour.	Petrol Consumption: Pints per B H P hour	B H P	Air, cubic feet per minute	Ratio. Air to petrol by weight	Per cent blower full	Exhaust Gas Analysis per cent by volume: CO_2	O_2	CO	N_2, &c.	Per cent excess of air.	Per cent loss of charge	ηP = mean pressure referred to B H P.
C. 10	400	—	—	0	—		—	6·4	6·5	4·3	82·8	19·8	20·5 (30)	—
B. 8	800	0·88	1·67	4·2	17·6	12·8	61	5·0	6·6	6·9	81·5	14·3	20·8 (30·6)	30·5
C. 11	1225	1·50	1·16	10·3	40	17	91	4·5	5·1	9·1	81·3	2·5	14·1 (23·2)	48·7
C. 13	1240	1·33	0·97	11·0	12	20	94	7·1	4·6	5·5	82·8	8·4	11·8 (20·8)	51·2
C. 13b	1240	2·0	1·46	11·0	42	13·4	94	—	—	—	—	—	—	—
C. 9	1225	1·48	1·02	11·5	40	17·3	91	5·5	5·1	7·0	82·4	7·3	14·1 (23·2)	54·2
B. 7	1820	2·12	1·41	12·0	—	—	—	3·3	6·1	6·8	83·8	12·3	18·6 (27·6)	38·2
B. 6	1560	1·58	1·0	12·6	56	22·5	100	6·4	5·5	5·2	82·9	13·2	15·9 (25·0)	46·7
B. 8b	1480	—	—	13·5	50	—	95	—	—	—	—	—	—	52·7
C. 12	1370	1·89	1·0	15·0	49·5	16·7	100	9·0	5·9	1·5	83·0	23·3	17·7 (26·7)	63·2
B. 7b	1970	—	—	15·1	—	—	—	—	—	—	—	—	—	44

square inch, a very good figure considering the size of the engine and the low ratio of the pump volume. The proportion of the fresh charge that escapes unburnt through the exhaust ports is given as 26·7 per cent, based on the assumption that the volumetric efficiency of the pump is 100 per cent, and that combustion is complete when the exhaust ports open.

If, as Mr. Brewer suggests, 2 per cent of oxygen is normally present in the exhaust gases of any petrol-engine, due to incomplete

combustion, then the loss of fresh charge is reduced to 17·7 per cent; and if, again, the volumetric efficiency of the pump be taken as 80, and not 100 per cent, as is more probable, then it is further reduced to 14·2 per cent. The fuel consumption, 1 pt. or 0·91 lb. per B.H.P. hour, is high, and the author believes that this is to be accounted for to some extent by precipitation of liquid petrol in the receiver. The thermal efficiency per B.H.P. hour, with petrol of 0·725 specific gravity and 18,500 B.T.U.s per lb., works out at 15·2 per cent a poor result. However, in the discussion upon Professor Watson's paper, read before the Institution of Automobile Engineers, Mr. Brewer stated that he had been able to obtain a brake thermal efficiency from this engine of 21 per cent when running at 1066 R.P.M., and developing 12·2 B.H.P. This corresponds to an indicated thermal efficiency of 28 per cent, assuming that the mechanical efficiency at this speed is 75 per cent. If the compression ratio be taken as 4·55 : 1, then the air-standard efficiency is $E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1} = 45{\cdot}5$ per cent, and the relative efficiency 61·5 per cent. If allowance be made for the loss of 14·2 per cent of unburnt mixture, then the brake and indicated efficiencies become 24·5 and 32·7 per cent respectively, and the relative efficiency 71·9 per cent.

Compared with the Lucas engine the Lamplough is (1) capable of maintaining a good mean pressure up to very much higher speeds; (2) it is much lighter; (3) the specific power from a given size of cylinder is much greater; (4) the rotative balance is almost equally good, but of course it does not possess reactionary balance; however, the higher speed and the greater number of impulses per minute makes this of less importance; (5) it is capable of firing regularly over a wider range of speed, and is less susceptible to variations in the density of the mixture.

The method of placing the cylinders across the crankshaft appeals to the author as somewhat unmechanical, in that it necessitates the use of bent connecting-rods, which, to be sufficiently strong, must necessarily be somewhat heavy, and, moreover, it increases the side-thrust of the pistons against the cylinder walls, due to the excessive angularity of these rods. So far as the author is aware, the Lamplough engine is not well known commercially, but it is, in some respects, one of the most interesting two-cycle engines yet produced.

The Dolphin Engine.—The engine illustrated in figs. 99 and 100 was designed by the author in conjunction with Mr. H. A.

Hetherington. In an earlier form it was placed on the market some nine years ago under the name of the Dolphin engine, but, owing to its inability to run at high speeds, and therefore to develop a high

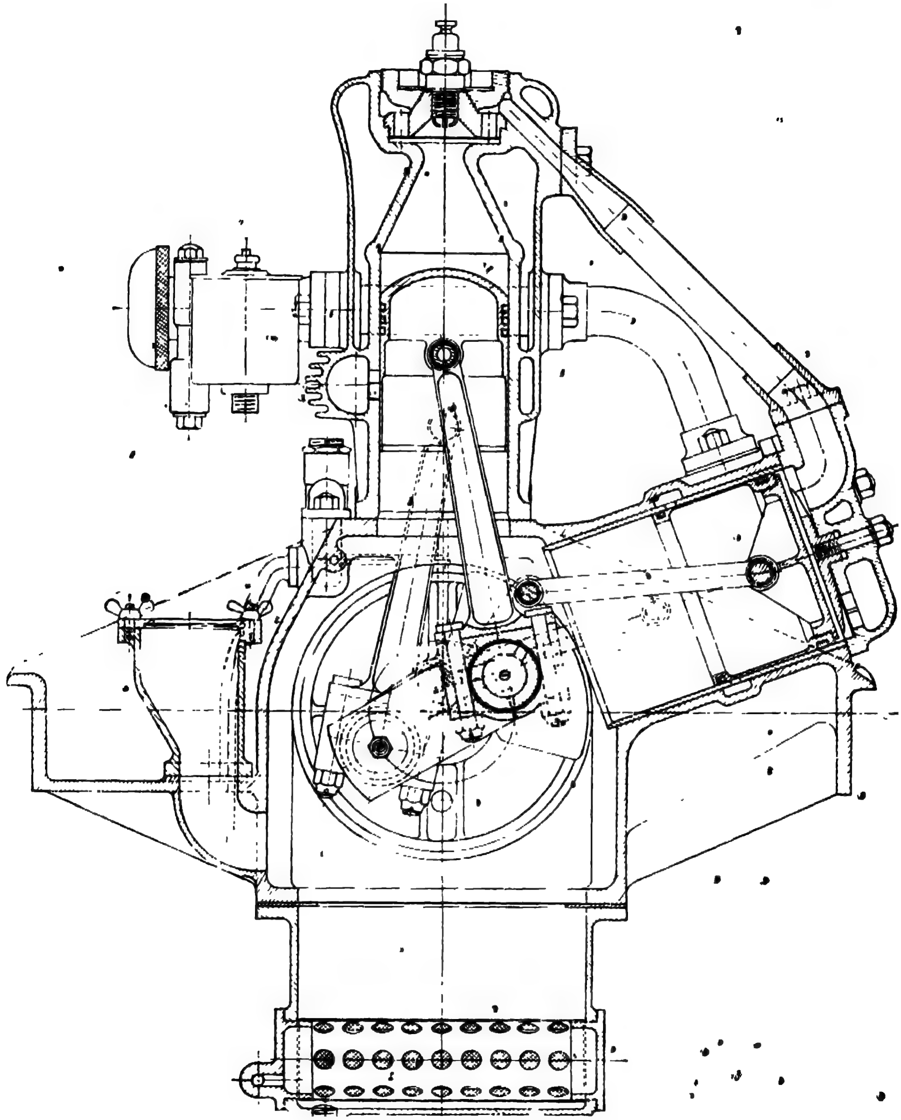


Fig. 99. General Arrangement of Two-cylinder 84-mm. Bore, 95-mm. Stroke. Type S.F.S.

specific power, and also to its low mechanical efficiency, it was unable to compete for automobile purposes with the four-cycle engine which at that time was making especially rapid progress. Recently, however, a very great improvement has been made in the valves, by

means of which it has become possible nearly to double the maximum rotational speed, and at the same time improve the mechanical efficiency. It will be seen from the illustration that the general lines of the engine are modelled upon the Clerk or Korting engines. While simplicity and cost of manufacture have been kept in view, neither efficiency nor balancing have been allowed to suffer on this

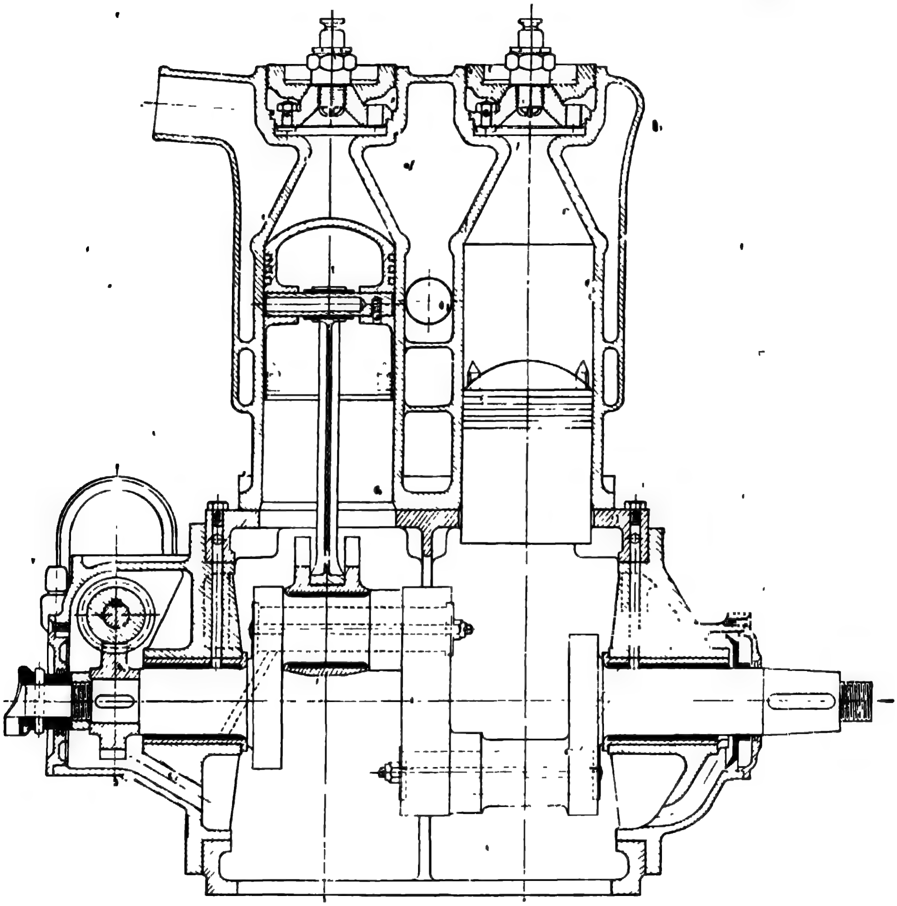


Fig. 100.—Ricardo Engine, Type S.F.S. Two-cylinder, 84 mm. × 95 mm.

account. The chief objects which have been aimed at are (1) ability to fire regularly over a very wide range of load and speed; (2) ability to attain a high mean pressure, and maintain it up to high rotational speeds; (3) good balance; (4) the suppression, as far as possible, of fluid and friction losses; (5) to obtain a high brake thermal efficiency at all loads.

To fulfil the five conditions mentioned above, the following means have been adopted. First, it was considered essential that stratifi-

cation should be encouraged by every possible means, and that the presence of a portion of undiluted working fluid in the neighbourhood of the igniter should be ensured at all loads and all speeds. The former was effected by so shaping the cylinder-head that the gases, after passing through the valves at high velocity, should be first collected into a small chamber and then passed into the cylinder, in the form of a solid diverging cone, at a low velocity, care being taken to ensure that no core of exhaust gases could be left in the centre of the cylinder. Reliability of ignition is effected by fitting the igniter in the centre of the small chamber just referred to, and so shaping the top of this chamber that a portion of the incoming charge is always deflected up against it; at the same time, the capacity of this small chamber is so proportioned in relation to that of the combustion space, that even on the lightest loads it is always filled with fresh combustible mixture. By this means there is always a readily ignitable charge in the ignition chamber.

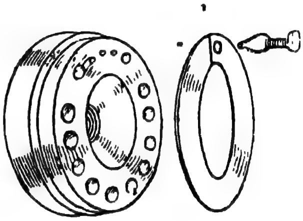


Fig. 101

The second condition, high mean pressure, has been met by careful attention to the scavenging, to ensure stratification, and hence the ability to admit a comparatively large charge of mixture without a serious proportion being lost through the exhaust ports. It is also helped by the employment of a special type of valve, which is capable of operating at exceedingly high speeds, and of dealing with a large quantity of mixture. This valve is illustrated separately in fig. 101, where it will be seen to consist of a split annular ring, of spring steel, held at one end, and free to lift off the seating at the other. Ports are cut in the seating round about 270 degrees of the circumference. The valve is, in effect, merely a flap valve, but, since it is exceedingly light, and is composed entirely of active spring material, its speed of working is very high, while the resistance it offers is small. In practice, about 10 oz. per square inch is required to fully open a valve which has a natural periodicity of 180 complete vibrations per second, permitting of an engine speed of well over 3000 R.P.M.

Prior to the adoption of these valves, the very highest speed which the author could obtain from an automatic poppet valve, with due regard to fluid losses, was about 50 periods per second, and to obtain this speed a spring tension equal to nearly 5 lb. per square inch was necessary, although the valve was made from nickel steel.

hollow, and lightened in every possible way. Fifty periods per second sufficed only for a speed of about 1000 to 1200 R.P.M., while the spring tension of 5 lb. per square inch involved very high pump losses. With the employment of this special valve the author

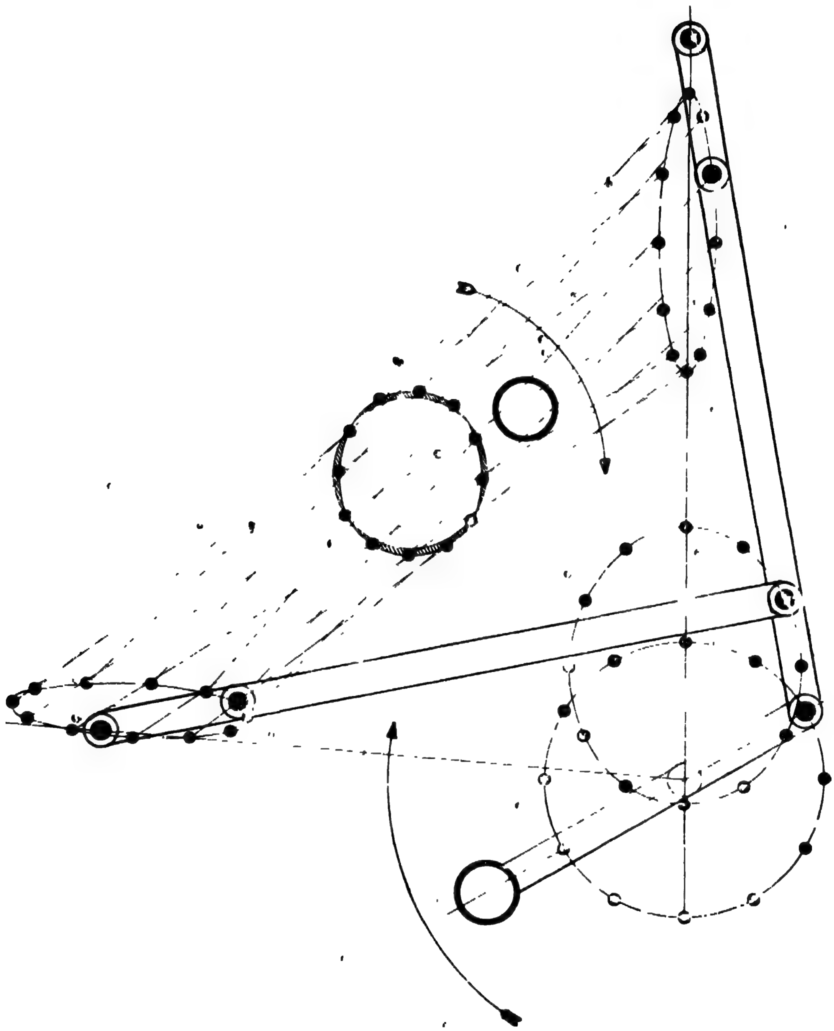


Fig. 102

has been able to maintain an indicated mean pressure of between 90 and 95 lb. per square inch at all speeds up to about 2000 R.P.M. in an engine of $3\frac{3}{4}$-in. stroke.

The third condition, good balance, is met by so arranging the relative motions of the pump and power pistons that the locus of their centre of gravity is almost a true circle, as shown in the

diagram (fig. 102), in which the dotted circle represents the path of the common centre of gravity of the two pistons. If, now, a balance weight equal to the weight of the two pistons and rods be fitted to the crankshaft in a position at 180 degrees to that of the common centre of gravity, and at a radius equal to that of the locus of the common centre of gravity, an almost perfect balance can be obtained with two or more units. The balance is not quite perfect, owing to the fact that the common centre of gravity does not describe a true circle, its path being slightly elliptical; this causes a slight disturbance, which occurs twice every revolution, but which is almost imperceptible. The balance actually obtained is very much better than that of a four-cylinder four-cycle engine of the ordinary type.

The fourth condition, low fluid and friction losses, is met by so timing the pump piston in relation to the power piston that nearly the whole of the delivery stroke of the pump takes place while the exhaust ports are uncovered. No receiver is employed, and the pressure in the pump cylinder scarcely exceeds that required to overcome the exhaust back-pressure and the slight resistance of the spring valve. No delivery valve is fitted to the pump, the author considering that the increased fluid resistance that such a valve would cause would more than counterbalance any advantage to be obtained from a reduction of clearance, which can always be met by slightly increasing the capacity of the pump. In order to reduce the clearance to a minimum, the pipe connecting the pump and working cylinders is of comparatively small bore, and the gas velocity through it is very high. Care is taken, however, that there shall be no sudden changes in the direction or velocity of the gases between the pump and power cylinders, such changes having been found both to increase the fluid friction and cause precipitation of the fuel.

The suction valve of the pump consists of a number of radial fingers of very thin sheet bronze, each finger covering a port. These fingers are held in place by means of a curved keep-plate, which prevents them from opening too far and being unduly strained, and also from local bending. The effective area of these suction valves is comparatively large, and their resistance trifling. However, in spite of the fact that the gas velocity through these valves is only 100 ft. per second at a speed of 2000 R.P.M., the volumetric efficiency of these pump cylinders, even at low speeds, is not as good as the author anticipated. Full-load indicator diagrams show considerable wire-drawing on the suction stroke, which is not easy to

account for; in some cases auxiliary ports have been fitted round the pump cylinders in such a position that they are uncovered by the displacer piston near the bottom of the stroke; these ports are in communication with the carburettor, and admit a further charge of combustible mixture. A large number of tests, taken from these engines, show that the fluid losses range from 5 to 7 per cent of the indicated horse-power at normal speed, and bear practically the same proportion at all loads.

The fifth condition, high thermal efficiency, is met by preventing diffusion between the incoming charge and the exhaust gases as far as possible, so that the highest mean pressure can be carried with the minimum loss of unburnt charge through the exhaust ports, and by the suppression of fluid and frictional losses. The latter can only be mitigated by careful attention to lubrication, and by reducing the weight of the reciprocating ports to the lowest limit. The author has not, up to the present, been able to reduce the fuel consumption below 0·67 lb. of petrol per B.H.P. hour, corresponding to a brake thermal efficiency of 20·5 per cent; but at half load the fuel consumption is almost exactly the same, and in this respect it compares very favourably with a modern four-cycle engine, in which the half-load consumption is generally from 15 to 20 per cent greater than the full load. For automobile purposes the light-load consumption is of more importance than the full load, and under such conditions this engine is probably quite equal to the four-cycle.

Referring to the sectional elevation, it will be seen that the two cylinders, together with their water-jacket and exhaust belt, are cast in one piece. The exhaust ports are arranged at equal intervals all round the cylinder wall, with the exception that one port is omitted on that side of the cylinder which receives the thrust from the connecting-rod. This is done in order to guard against the escape of lubricant through the port at the point where it is most needed. The ports are not very large, having an area equal to only about 15 per cent of the area of piston, and their height is such that they are first uncovered when the piston is 55 degrees before the bottom centre. It was found that larger exhaust ports, owing to the very sudden release of pressure in the cylinder, set up violent eddies in the exhaust gases, and caused considerable diffusion. Reducing the area of the ports had the effect of greatly improving the scavenging without apparently raising the exhaust back-pressure to any measurable extent.

In matters of this kind the designer is faced with the problem that these small engines are called upon to run not only at varying loads, but also at very widely varying speeds, and he has to decide what is the most probable speed and load at which the engine will be running during the greater part of its existence, and then design the exhaust ports, &c., for that speed. Examining the shape of the combustion chamber, it will be observed that the surface exposed to combustion is large. From a thermodynamic point of view this is undoubtedly wrong, and it is probable that an appreciable amount of heat loss might have been saved, even at high speeds, had a more compact chamber been used. In this case, however, the author considered it of much greater importance to satisfy the conditions required for good scavenging, even at the expense of some extra heat loss. The top of the cylinder is closed by the valve seat, which thus forms the cylinder-head, and is held in position by means of a heavy, gun-metal, screwed plug, whose function it is to conduct heat away from the centre of the seating to the cooling-water. In order that the transfer of heat shall be as rapid as possible, the bearing surface between the seating and the plug is in the form of a wide cone, both surfaces being ground to ensure good contact.

Both the valve and the igniter are attached to the valve seat, and all three can be withdrawn for inspection and cleaning by merely unscrewing the plug. A shoulder is provided in the cylinder, upon which the valve seat bears, and here again a metal-to-metal ground joint is employed, in order to assist in the transference of heat. Below this shoulder is a second and wider one which acts as a stop for the valve and limits its lift. The combustible mixture from the pump cylinder enters between the screwed plug and the seating, and, passing through the ports of the latter, lifts the valve and enters the small igniting chamber at a high velocity. The gases enter around nearly the whole inside circumference of the valve, and converge at the centre. Thus they first fill the small ignition chamber and then pass through the constricted neck into the cylinder. The conditions governing good scavenging and rapid and complete combustion appear at first sight to be conflicting, for the former requires that the gases shall enter the cylinder at a low velocity, and free from eddies, &c., which may cause diffusion, while the latter requires that they shall be in a state of rapid motion at the time when ignition occurs. In this case, since ignition takes place primarily in the small ignition chamber, where the gases are probably still in motion, the sudden expansion of the

charge therein causes a violent disturbance in the main body of the gases in the lower part of the combustion space, and so serves to promote rapid combustion. Indicator diagrams taken at high speeds show that the combustion is extremely rapid, and that with the spark timed to take place 15 degrees before the dead centre, the maximum pressure is attained when the piston has travelled through only 2 per cent of its stroke.

The inlet valve itself is made of spring steel, hardened, tempered, and ground, the thickness being about ·05 in., but varying somewhat according to size and speed. Since, during the compression and expansion stroke, this valve is in good contact throughout the whole of its surface with the seating, it follows that, no matter what the temperature be in the cylinder, the temperature of the valve can only be very slightly higher than that of the seating, and can easily be kept within safe limits; furthermore, the incoming charge passes over the valve at each revolution and assists in cooling it. In practice this arrangement of valve, seating, and plug has been found to work satisfactorily for all sizes up to 3½-in. valve diameter, but in larger diameters than this the temperature of the centre parts of the valve seating and of the igniter is liable to become so high as to cause pre-ignition.

The ratio between the swept volume of the pump and that of the working cylinders ranges from 1·5 : 1 in the smaller sizes to 1·21 : 1 in the larger. In comparing these figures with the Lamplough or other similar engines, however, it must be remembered that the clearance loss accounts for about 25 per cent of the pump capacity, so that the actual ratios are only about 1·14 : 1 and 0·90 : 1 respectively.

The compression ratio is approximately 4 : 1 for all sizes when petrol is employed, and 5·5 : 1 for town gas, so that the ratio of the effective pump capacity to the total cylinder capacity is about 0·84 : 1 and 0·64 : 1 respectively for petrol. With these ratios, indicated mean pressures of 95 and 80 lb. per square inch are generally obtained. The remainder of the engine calls for little comment. Forced lubrication is provided to all bearings by means of a rotary oil pump driven by spiral gearing from the crankshaft. The main bearings are lined with white metal. Until recently, the author invariably used ball-bearings for the crankshaft, but the strenuous competition for silent running led him very reluctantly to abandon their use.

The engine illustrated in figs. 103 and 104 is of the stationary

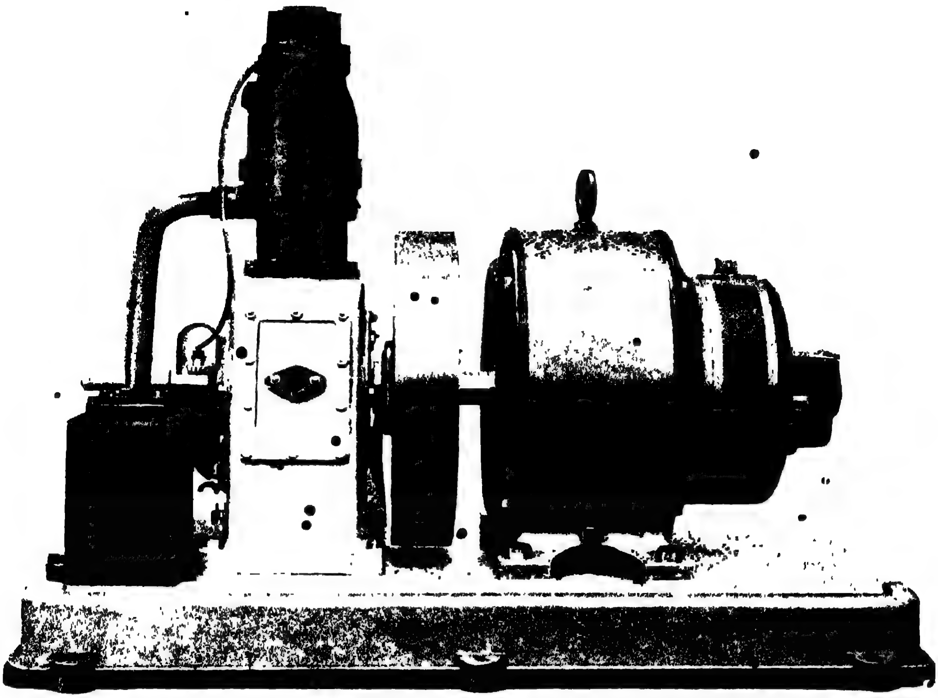


Fig. 103 - 7·5-B.H.P. Single-cylinder Stationary Dolphin Engine

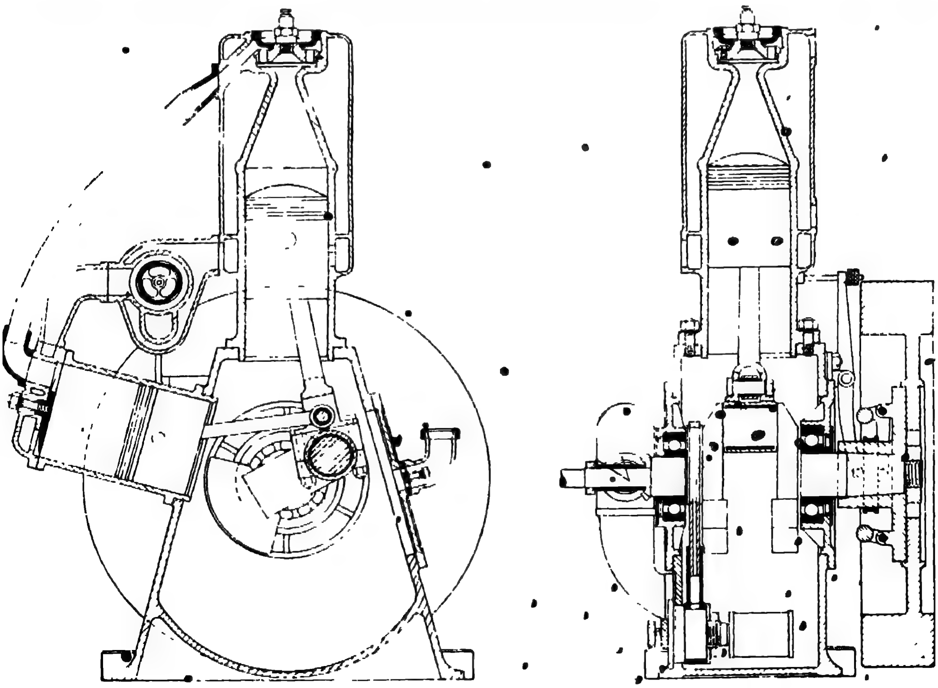
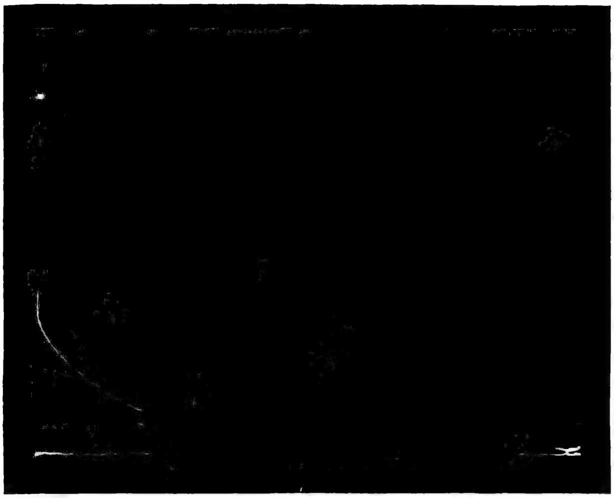


Fig. 104 —Sections of 7·5-B.H.P. Stationary Engine

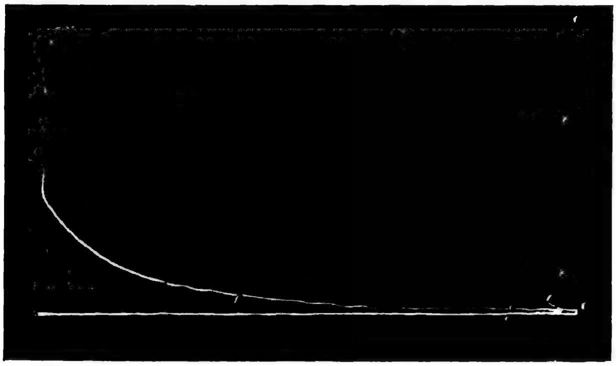
type, and is mainly used for the electric lighting of private houses or yachts. It can be arranged to run on either petrol or gas, the only difference being that with the latter fuel a higher compression is employed, and the carburettor is replaced by a mixing valve. The engine has a bore of $4\frac{1}{2}$ in. and a stroke of $5\frac{1}{2}$ in., and is rated at 7·5 B.H.P. when using petrol and running at 700 R.P.M. At this speed it is, however, capable of developing up to $9\frac{1}{2}$ B.H.P. as an overload. This corresponds to an ηp value of 61·5 lb. per square inch, the mechanical efficiency is 81 per cent, and the indicated mean pressure 76 lb. per square inch. The best fuel consumption is about 0·72 lb. per B.H.P. hour, corresponding to a brake thermal efficiency of 19 per cent. The swept volume of the power cylinder is 88 cu. in., and that of the pump cylinder 112 cu. in. Allowing for 25 per cent clearance loss, the ratio between the swept volumes is as 0·95 : 1, and as compared with the total volume of the working cylinder, it is as 0·7 : 1. With this ratio a mean pressure higher than 76 lb. per square inch should be obtainable, but indicator diagrams taken from the pump cylinders of these engines show a considerable amount of wire-drawing on the suction stroke, which reduces the volumetric efficiency of the pump, and is not readily accounted for.

The indicator diagrams illustrated in figs. 105, 106, and 107 were taken from one of these engines with a Hopkinson optical indicator. At the time when the diagrams were obtained the engine had been in continuous service for about nine months, driving a direct-current generator, and running on the average about 14 hr. per week. For the power cylinder, a spring giving 90 lb. to the inch was used in the indicator, and for the pump cylinder a 20-lb. spring. Diagrams Nos. 1 and 2 were taken from the power and pump cylinders respectively, when the engine was running at slightly above its normal load, or 7·9 B.H.P., Nos. 3 and 4 when the load was reduced to 5·5 B.H.P., Nos. 5 and 6 at 3·8 B.H.P., and Nos. 7 and 8 at no load. In this latter case a 20-lb. spring was used in the indicator for the power cylinder also. Diagram No. 9 was taken from the pump cylinder, with an overload of 23 per cent, or 9·2 B.H.P., but no diagram could be obtained from the power cylinder under these conditions, owing to the seizure of the indicator piston. Repeated attempts were made, but, although the diagram could be examined, the indicator could not be kept running for a long enough period to enable a photograph to be taken, without overheating. The time of exposure varied somewhat, but in no case was it less than $\frac{1}{3}$ second, during which time the engine made slightly over



No. 1

No. 2



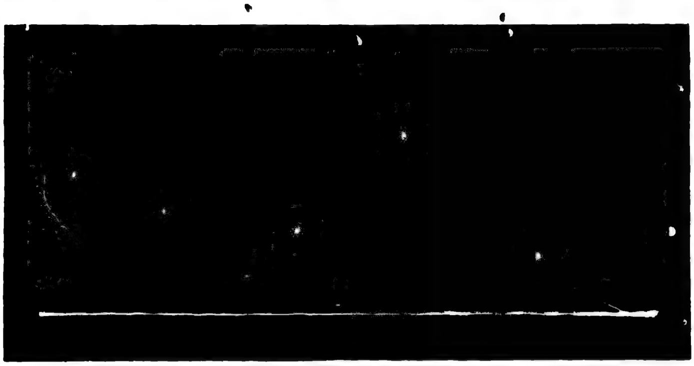
No. 3

Fig. 105

two revolutions. In the case of the light-load diagram the exposure lasted over one second, and all the pump diagrams were taken with a still longer exposure.

Since, so far as the author is aware, these and the diagrams taken by Professor Watson on a Day engine, which will be discussed later, are the only ones that have so far been taken from a small high-speed two-cycle engine, it is, therefore, worth while to examine them in some detail. In the first place, it will be noted that the

No. 4

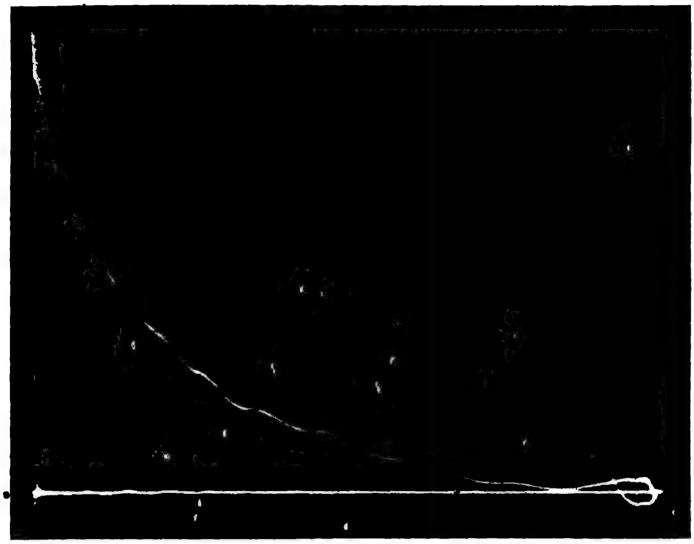


No. 5

No. 6

Fig. 106

compression-pressure varies very considerably with the load; this may be attributed to two distinct causes: (1) with increase of load the pressure in the cylinder, at the moment when the ports are closed, is greater, due, of course, to the greater exhaust back-pressure, but this alone does not account for the very great difference between the pressure of compression at no load and full



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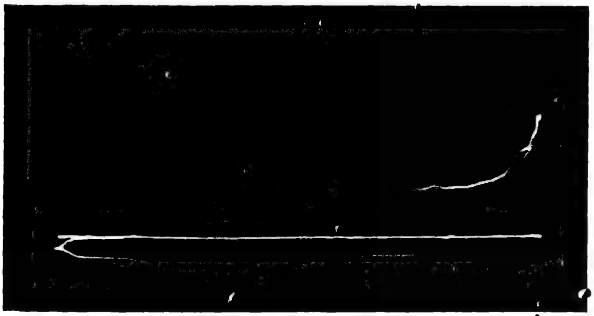


No. 9

Fig. 107

load; (2) the second cause is that on light loads the proportion of exhaust gases retained in the cylinder is very much greater, and the temperatures of these gases is so high that they are still giving up heat to the walls of the cylinder during the compression stroke, at as great, or even a greater rate, than it is being imparted

to them by the compression. That is to say, the compression curve on light loads, instead of being adiabatic, is practically isothermal, and the temperature of the contents of the cylinder at the end of compression is little or no higher than at the beginning. Another feature of these diagrams is that even with a mean pressure of only 35 lb. per square inch, the rise of pressure on combustion is extremely rapid, showing that the mixture around the plug must be fairly pure, for any serious amount of dilution due to diffusion would tend to retard combustion. In this connection it should be added that ignition took place in all cases 10 degrees before the top dead centre.

The rapidity of combustion at light loads, when the proportion of fresh gases retained in the cylinder cannot exceed 10 per cent of the cylinder volume, and is probably even less, shows fairly conclusively that a considerable amount of stratification does take place. For, if it were supposed that the 10 per cent of fresh gases were intimately mixed with some 90 per cent by volume of exhaust products, it is clear that combustion would not take place at all. The author must apologize for labouring this point; but, since it is a question upon which the leading authorities are not in agreement, and upon which the whole working process of two cycle engines largely depends, it is worth while devoting a good deal of space to it.

The opening of the exhaust ports and the rapidity of the pressure drop are very clearly shown in each of the diagrams. It will also be observed that the pressure in the cylinder drops to its lowest value when the piston is about 5 per cent before the end of the stroke, showing that the area of the exhaust ports is ample for the speed at which the engine is running. It is clear from these diagrams that if bottom scavenging were employed, without a delaying valve, it would be necessary to make the exhaust ports open 15 per cent of the stroke before the inlet ports. Even if the depth of the inlet ports were only 15 per cent of the stroke, which would involve a very high velocity through them, and therefore considerable fluid friction, the exhaust ports would still have to be open somewhere about 30 per cent before the end of the expansion stroke, instead of 20 per cent as in this case.

In the no-load diagram it will be observed that the exhaust pressure actually drops below atmospheric. To obtain this result, the governor was cut out of action, and the engine speeded up until the pressure oscillations in the exhaust pipe synchronized with the

opening of the exhaust. It is, however, just possible that the loop shown is due to the inertia of the indicator piston, though the author does not think that this is probable, for three reasons: (1) The oscillation does not coincide with the natural periodicity of the indicator; (2) the pump diagrams do not show any evidence of oscillations being set up in the indicator when the atmospheric line is crossed on the suction stroke (a similar condition); (3) the pressure in the pump cylinder (see pump diagram, fig. 107, No. 8) remains below atmospheric, and only crosses the atmospheric line at the moment when the exhaust ports are closed.

Referring to the pump diagrams, it will be observed that the pressure rises rapidly at the end of the delivery stroke, due to the fact that this portion of the stroke is performed after the closing of the exhaust ports by the power piston, and the gas is therefore compressed into the transfer pipe, and then re-expanded. The suction line crosses the atmospheric line again about 25 per cent after the beginning of the suction stroke. From this it will be seen that the last 25 per cent of the pump's stroke is devoted to compressing the gases into the clearance space and re-expanding them. This portion must, therefore, be written off as clearance loss. A considerable portion of this loss could be avoided if delivery valves were fitted between the pump cylinder and the transfer pipe.

The point at which the inlet valve of the power cylinder opens is particularly well shown on the overload diagram, and the oscillations on the delivery stroke of this diagram correspond with the natural periodicity of the spring valve. The diagrams also show very clearly the great advantage that is to be gained by the use of an automatic valve, which does not open until the pressure in the power cylinder has fallen below that in the pump. If a mechanically operated inlet valve were fitted, and timed to open at the correct point for full-load running, then on light loads it would be opening at a time when the pressure in the pump was considerably below atmospheric, with the result that the products of combustion would be driven back into the pump cylinder, fouling, and possibly igniting, the charge therein.

Firing-back into the pumps or receivers is liable to occur with all two-cycle engines which do not use a primary air-scavenge. If the capacity of the receiver is large, such back-fires are serious in that they destroy a large quantity of combustible mixture, and for the next few strokes the engine receives fouled charges, which either do not ignite, or burn so slowly as to increase the risk of firing-back

again. In engines such as the one under consideration, in which no receiver is employed, such back-fires as do occur are of little consequence, in that they affect only that charge which is entering at the time, and have little or no effect on the subsequent charges.

The curves shown in fig. 108 are taken from a long series of tests carried out on a small two-cylinder engine of a type suitable for motor-car work.

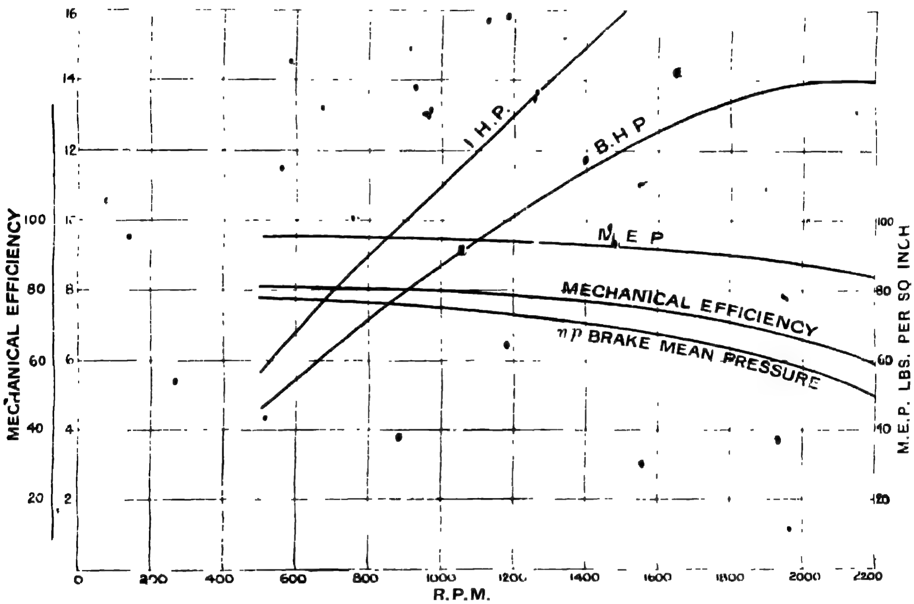


Fig. 108

The leading dimensions of this engine were as follows:—

Power cylinders bore	2·8 in.
Power cylinders stroke	3·75 in.
Swept volume	23 cu. in.
Compression ratio	3·9 : 1.
Total volume	31 cu. in.
Area of exhaust ports	1·5 sq. in.
Exhaust ports open (degrees)	55°.
Percentage of stroke	18 per cent.
Effective area of inlet valve	0·9 sq. in.
Pump cylinder bore	4 in.
Stroke	2·8 in.
Swept volume	35 cu. in.
Effective volume	26·3 cu. in.
Effective area of pump inlet valve	1·4 sq. in.
Ratio swept volume of pump to swept volume of pump cylinder	1·51 : 1.
Ratio effective volume of pump to total volume of power cylinder	0·84 : 1.

From examination of the curves, it will be observed that the B.H.P. increases with increase of speed up to nearly 2200 R.P.M., corresponding to a piston speed of 1370 ft. per minute. It is thus clear that up to this speed the inlet valve is operating quite satisfactorily. Above 2200 R.P.M. the mean pressure begins to fall more rapidly, and the indicated horse-power no longer rises in a straight line. This is probably due partly to wire-drawing in the pump valves, and partly to excessive exhaust pressure, which, by raising the pump pressure, will increase the clearance losses. The mean effective pressure up to 1800 R.P.M. exceeds 90 lb., reaching a maximum of 95·5 lb. per square inch at 800 R.P.M. The B.H.P. reaches a maximum of 14 at about 2100 to 2200 R.P.M., but above that speed it falls off.

The mechanical efficiency, as might be expected, falls steadily with increase of speed, due to the extra fluid and friction losses, both of which will increase nearly as the square of the speed. At 500 R.P.M. the mechanical efficiency is 82 per cent, at 1000 R.P.M. 80 per cent, at 1500 R.P.M. 76 per cent, and at 2000 R.P.M. 66 per cent. The fuel consumption at full load was found to be approximately 0·7 lb. per B.H.P. hour, and at half load 0·72, the speed being 1480 R.P.M. in both cases. This consumption corresponds to a brake thermal efficiency of 19·7 and 19·1 per cent respectively, the indicated thermal efficiency on the full-load test being 26·0 per cent, and on the half load 31·0 per cent. The air standard efficiency for this engine is approximately 42·2 per cent, so that the relative efficiency is 61·6 and 73·5 per cent respectively at full and half loads; the high relative efficiency at half load being, of course, due to the system of qualitative governing rendered possible by the high degree of stratification obtained. The lower thermal efficiency at full load is evidently due in part to the loss of a portion of the unburnt gases through the exhaust ports, which is only to be expected in an engine with so large a pump ratio. In such a case as this, the full capacity of the pump is only called for under abnormal conditions, as when climbing a steep hill, and for the greater part of the engine's life it is running at half load or less.

CHAPTER XX

LOW-EFFICIENCY TWO-STROKE ENGINES

So far we have considered only those two-cycle engines which may be described as high-efficiency engines, that is to say, engines in which good balance, high specific power, flexibility of both torque and speed, and low fuel consumption have been the chief objects in view. There is a still larger class of two-cycle engines, in which balance, specific power, and fuel consumption are sacrificed in favour of extreme simplicity and low first cost. For such engines, especially in small powers, there is undoubtedly a very large market. They are, for example, especially suitable for auxiliary engines for sailing yachts, dinghies, motor cycles, agricultural purposes, &c., and for all purposes where low first cost is the most important consideration, where, owing to the short period that they are required to work, the fuel consumption is not a serious item, and when they are not called upon to run at widely varying speeds or under widely varying loads. For all such purposes the two-cycle engine using crankcase compression and scavenging with combustible mixture has a wide scope, for it is probably the simplest form of prime mover in existence, has very few parts, and is exceedingly cheap to produce. In America, in particular, such engines are produced in enormous quantities, in sizes ranging from 1 to 12 horse-power per cylinder. They are also, in exceptional cases, built in larger sizes, but their excessive fuel consumption and other inherent disadvantages generally put them out of court. By far their largest field is in the propulsion of yachts' dinghies, and they are frequently made in the form of a detachable "motor rudder", a purpose for which they are admirably suited.

Although by far the larger proportion of small two-cycle engines at present on the market are of the crankcase compression type, the author does not propose to devote a great deal of space to discussion of the various makes. In the first place, the difference between them is very slight, and their inherent disabilities will prevent them from competing with the four-cycle engine, or the high-

efficiency two-cycle, on any grounds except first cost and mechanical simplicity. The word mechanical should be underlined, because, in practice, they are anything but simple to handle, owing to their tendency to back-fire and reverse on the slightest provocation.

Tests of a Day Engine.—So far as the author is aware, although a vast number of these little engines have been built, very few scientific tests have been carried out on them. Professor Watson, however, has carried out a series of very detailed tests on

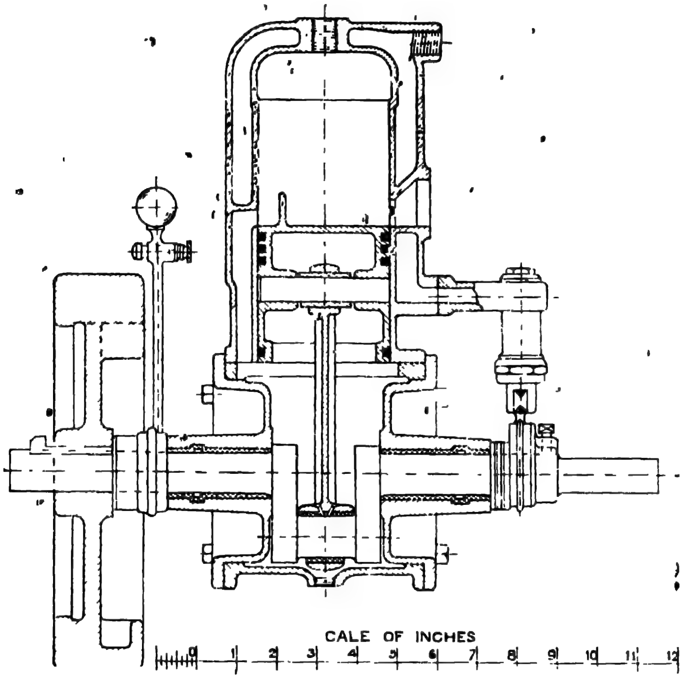


Fig. 109. Three-port Day Engine

a smal Day engine, rated at $2\frac{1}{2}$ B.H.P. These tests have already been referred to when discussing the question of scavenging. The engine he experimented with is of the three-port type, and may be regarded as typical of engines of this class. A cross-section of it is shown in fig. 109, which shows the relative positions of the inlet and exhaust ports in the cylinder, but does not show the inlet port to the crankcase, which, in the position illustrated, is masked by the piston. The port timing is shown in fig. 110. The exhaust port is open during 122 degrees of the crank, and the inlet port during 97 degrees, the "lead" of the exhaust port over the inlet being $12\frac{1}{2}$ degrees. The inlet port to the crankcase opened for 69 degrees as the engine was originally constructed, but Professor Watson found

that a considerable improvement could be made to the power of the engine at high speeds by increasing the depth of this port, so that it was opened for 82 degrees. The leading dimensions of the engine were as follows:

Bore ...	3·25 in.
Stroke ...	3·25 in.
Swept volume ..	27 cu. in.
Total volume ...	34·5 cu. in.
Compression ratio	4·67 : 1.
Volume swept by pump	27 cu. in.
Area of ports. ...	Not given.

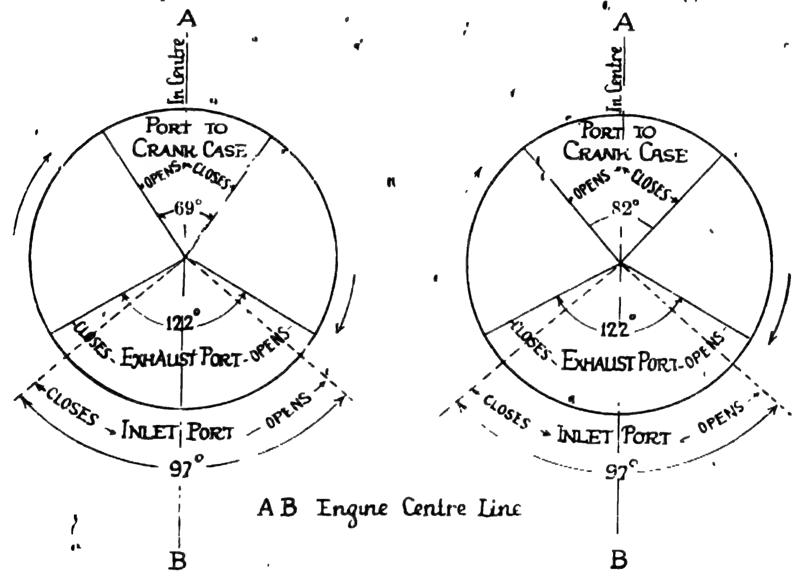


Fig 110.—Valve Diagrams

Tests were carried out, and indicator diagrams taken, at 600, 900, 1200, and 1500 R.P.M. The diagrams are illustrated in figs. 111 and 112, and the port openings in each case are marked on the diagram, so that the pressure, both in the cylinder and crank-case, at the time of opening and closing the ports, can be read off.

The results are tabulated below:—

Test No	Speed.	B.H.P.	I.H.P.	M.E.P.	Air Petrol Ratio	Mechanical Efficiency.	ηp	Friction Loss	Fluid Loss
						Per cent		Per cent.	Per cent.
4	600	1·9	2·48	59·7	13·88 : 1	76	45·5	16·0	8·0
1	636	2·1	2·71	62·7	11·21 : 1	77	48·5	15·3	7·7
15	907	2·5	3·30	53·4	14·85 : 1	75	10·0	17·8	7·2
11	938	2·9	3·68	57·8	11·76 : 1	78·5	45·2	14·7	6·8
21	1212	3·3	4·29	52·0	13·87 : 1	77·0	40·0	16·25	6·75
20	1218	3·3	4·29	51·8	10·96 : 1	77·0	39·8	16·25	6·75
33	1508	3·6	4·91	47·9	13·98 : 1	73·5	35·2	20·45	6·05
26	1510	3·5	4·75	46·2	10·68 : 1	74·0	34·8	19·85	6·15

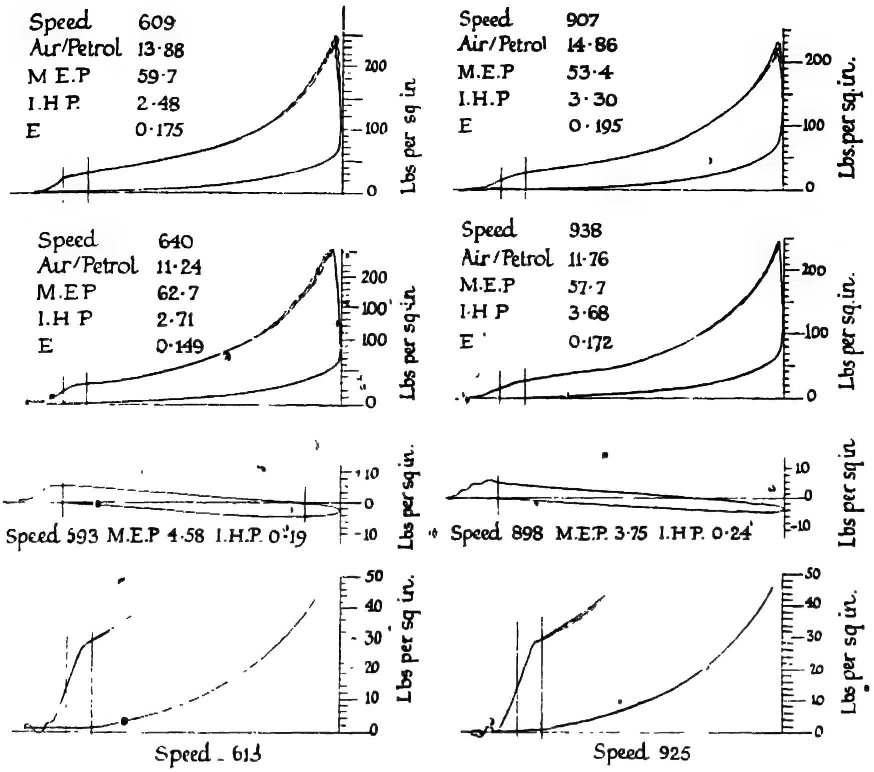


Fig. 111.—Indicator Diagrams from Day Engine

The fuel consumption and brake and indicated thermal efficiencies obtained during these tests were as follows:

Test No.	B.H.P.	I.H.P.	Lbs. per B.H.P. Hour	Lbs. per I.H.P. Hour	Brake Thermal Efficiency	Indicated Thermal Efficiency
					Per cent.	Per cent.
4	1·9	2·48	1·63	0·784	13·3	17·4
1	2·1	2·71	1·19	0·917	11·5	14·9
15	2·5	3·30	0·95	0·702	14·3	19·5
11	2·9	3·68	1·02	0·795	13·4	17·2
21	3·3	4·29	0·875	0·671	15·5	20·4
20	3·3	4·29	1·10	0·847	12·4	16·1
33	3·6	4·91	0·82	0·598	16·6	22·9
26	3·5	4·75	1·09	0·808	12·5	16·9

The author has selected only a few representative examples of the thirty-five tests published by Professor Watson. Test 33 is by far the best of any of them. The highest B.H.P. recorded is 3·70 at 1500 R.P.M., while at 1199 3·4 B.H.P. was developed. Since an increase of 300 R.P.M. gives a corresponding increase of only

0·3 B.H.P., it would appear that the maximum horse-power is reached at about 1500 R.P.M.

By dint of further experimenting with the ports and increasing the crank-chamber inlet port opening, as explained before, Professor Watson was able to increase the power at all speeds above 600 R.P.M., but this improvement was only obtained at the expense of the thermal efficiency. The improved results are given below:

Test No	Speed.	I.H.P.	B.H.P.	Mechanical Efficiency.	Brake Thermal Efficiency.	Indicated Thermal Efficiency.	p (lbs. per square inch).
				Per cent.	Per cent.	Per cent.	Per cent.
42	606	2·46	1·83	74·5	9·5	12·8	45·4
45	922	4·11	3·34	81·0	11·8	14·6	53·0
46	1200	5·18	4·20	81·0	12·5	15·4	51·3
47	1500	5·74	4·64	81·0	13·4	16·3	45·5

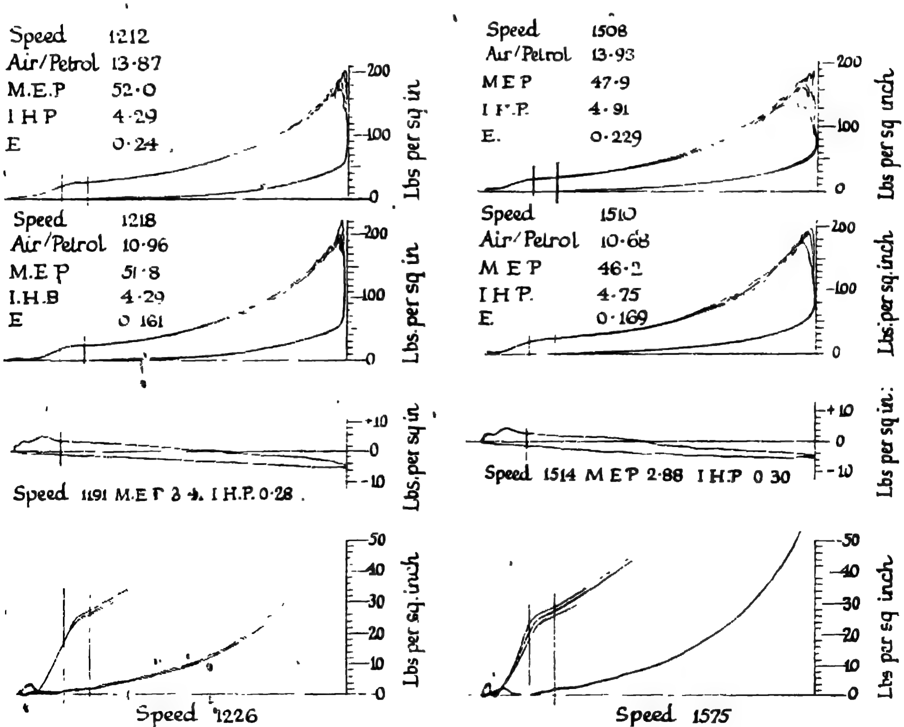


Fig 112.—Indicator Diagrams from Day Engine

The effect of this change has been to raise the specific power, and therefore also the mechanical efficiency, very considerably, but the thermal efficiency has been much reduced, owing to the larger proportion of fresh charge lost through the exhaust ports. The percentage of unburnt charge in the exhaust is as follows:—

Speed:	Original Inlet Ports.	Altered Inlet Ports
600	35 per cent	32·5 per cent.
900	29 ,,	34·0 ,,
1200	19 ,,	27·0 ,,
1500	7 ,,	17·0 ,,

From the general slope of the B.H.P. curve it is probable that the maximum B.H.P. would be about 4·8 at 1700 to 1800 R.P.M.

One of the greatest objections to engines of this type is their inability to run at light loads. This will readily be understood when it is considered that, even when running on full load, the proportion of combustible mixture to exhaust gases present in the cylinder barely exceeds 30 per cent, and that, owing to the violent diffusion set up by this system of scavenging, this small proportion is intimately mixed with the products of combustion. If, now, the proportion of combustible mixture be still further reduced, it becomes so diluted with exhaust products as to cease to be inflammable. In this case the engine ignites only every second revolution, that is to say, only after two charges have entered the cylinder. This is generally known as "four-cycling", and is a common and tiresome habit with engines of this type.

The following remarks may be quoted from Professor Watson's admirable paper on the Day engine, read before the Institution of Automobile Engineers:

"When comparing the working of this two-cycle engine with an ordinary four-cycle engine, it is to be noted that the range of mixture richness which it is possible to use is considerably smaller with the two-cycle than with the four-cycle, due to the very much larger admixture of exhaust products with the fresh charge. Unless the richness of mixture be adjusted within comparatively narrow limits, particularly at the high speeds, the engine refuses to work on the two-cycle, and only fires on every other out-stroke, the intermediate stroke acting as a scavenging stroke. The result of this peculiarity is, that unless the carburettor provides a mixture of uniform richness at different speeds and for different throttle openings, satisfactory working cannot be obtained. In the case of the engine under test, the effective size of the carburettor jet was hand-adjusted in every case, but even then, at a speed of 1500 R.P.M., it was often difficult exactly to hit off the correct mixture."

Referring to the retarding of combustion by the presence of so large a proportion of exhaust gases, Professor Watson says:

"The proportion of exhaust products to new charge is about

twice as great in the case of this two-cycle engine as it is in the four-cycle engine. By determining the crank angle at which contact was made on the commutator, and the interval which elapses, with the particular coil employed, between the closing of the primary circuit and the passage of a spark, it was found that to obtain the best-shaped diagram at 1500 R.P.M. the spark had to pass, and hence the charge was fired at, a crank angle of 30 degrees before the top of the stroke."

As regards mechanical features, it must be remembered that the chief claim of these engines to popularity lies in their low initial cost, and therefore, their mechanical design is carried out with a view to reducing cost of production to the lowest possible limit. It is common practice to cast the cylinder, together with its water-jacket and the main body of the crankcase, in one piece, leaving the top of the cylinder open, and afterwards closing it with a plain flat cover, often unjacketed. In engines of the better quality the cylinder is cast separately from the crankcase but in one piece with the head, the head in this case being hemispherical and water-jacketed. In all cases the ports are cored into the cylinder casting, and are not machined. The crankcase generally consists of a plain cylindrical body, provided with the necessary lugs for supporting the engine, and closed by two plain circular covers which carry the main bearings. Gas tightness in the crankcase is generally secured by the use of very long main bearings, lubricated with thick grease, an arrangement which works well so long as the bearing is not badly worn.

In the larger engines, in cases where space does not permit of very long bearings, sealing washers are employed. These consist of faced washers, fitted very carefully to the crankshaft, and made to revolve with it by means of a feather or pin; light springs are fitted behind these washers to keep them pressed against the inside walls of the crankcase, which are also carefully machined for their reception. Absolute tightness of the crankcase is a matter of considerable importance, for although the pressures are low, leakage of only a very small proportion of air into the crankcase will alter the composition of the combustible mixture therein, and may lead to incessant trouble. In the design of piston-head there is a great difference of opinion. Some designers shape the top of the piston with the greatest care, so that it shall deflect the incoming charge up to the top of the cylinder; others again, as in the Day, use a plain flat-topped piston, with nothing but a vertical deflector plate—

a design which seems highly unscientific, but in practice appears to give equally good results. In the large Sulzer-Diesel engines, using bottom scavenging, a perfectly plain piston is used, with a slightly concave head, and reliance is placed on the upward slope given to the ports. In practice these engines give the highest mean pressures of any that employ bottom scavenging; there are, however, other reasons which exert a powerful influence in this case. The author is inclined to believe that the use of baffles on the piston is unnecessary; there is probably a better chance of propelling the incoming charge up to the top of the cylinder by using upward sloping ports and giving the direction at a time when velocity of the gases is very high, than by attempting to deflect them upwards after they have entered the cylinder and lost much of their velocity.

With regard to the "lead" of the exhaust ports, there is again a considerable difference of opinion, and designers appear, through lack of published data, to be very much in the dark. An examination of the indicator diagrams taken from the Day engine reveals the fact that the lead given, in this case, is not sufficient except for quite low speeds; for the pump diagrams show a marked rise of pressure in the crank-chamber when the exhaust ports are first opened, showing that the pressure in the cylinder has not fallen sufficiently. The port areas in this instance are not given, so it is not possible to calculate the area of exhaust port that is uncovered before the inlet commences to open. To give the lead in terms of degrees of crank angle alone is not sufficient, the area also must be given before any useful data can be arrived at. It is evident that the success or otherwise of engines of this type must depend very largely on the exact proportioning and timing of the port openings, and that the very wide variations of output from apparently similar engines of different makes must be due to slight differences in the size or timing of these openings. The remaining parts of these engines call for little comment, since they do not differ in any way from similar parts of other engines, except that the crankcase is made as small as possible, and all clearances cut down to the minimum in order to reduce its capacity. Even so, as has already been pointed out, the clearances are far too great to permit of efficient pumping.

The Gray Motor.—In fig. 113 is shown an engine made by the Gray Motor Company, of Detroit, Michigan, U.S.A., who manufacture these engines in enormous quantities, their principal market being for marine work. As will be seen from the illustration, the

engine is of the three-port type, supplemented by a small auxiliary inlet valve. On the upward stroke of the piston the small port on the left-hand side of the cylinder is first uncovered by the piston, and combustible mixture is drawn into the crankcase through a spring-loaded poppet valve. Owing to the spring tension required to overcome the inertia of this valve at high speeds, the admission of the charge is necessarily somewhat throttled, and to obviate this an additional port is arranged, which is uncovered by the skirt of the piston when it has almost reached the end of its stroke. This port allows combustible mixture to flow freely into the crankcase and fill it; at the same time it allows the small inlet valve ample time to close. On the downward stroke of the piston the gas and air are entrapped in the crankcase and compressed.

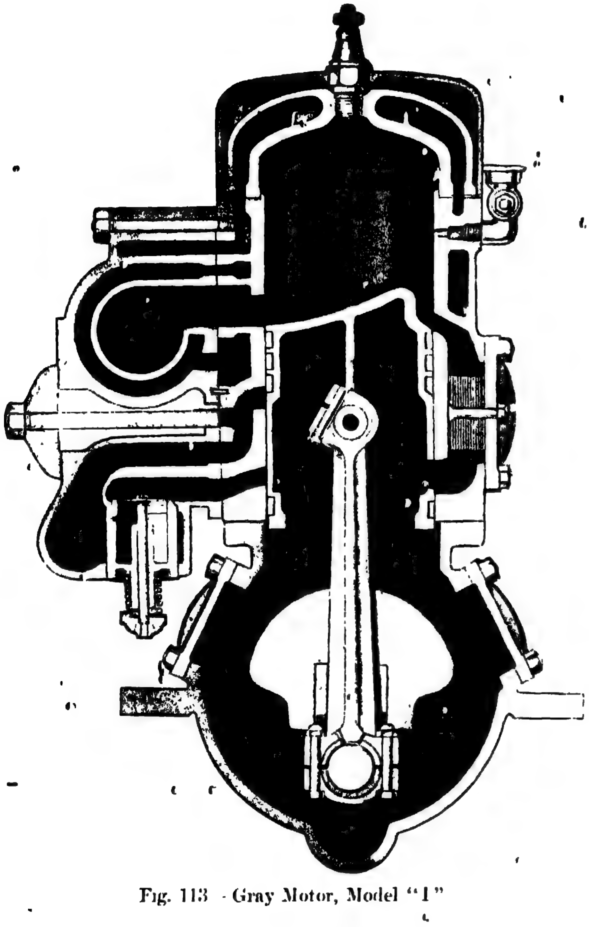


Fig. 113 - Gray Motor, Model "1"

The exhaust port on the left-hand side of the engine is uncovered by the piston when about 75 per cent of its stroke has been completed, and the inlet port at about 85 per cent of the stroke. It will be noted that the inlet ports are given a slight upward inclination, and that the baffle on the head of the piston is fitted very close to the inlet side of the cylinder, in order that the deflector shall act upon the gases at a time when their velocity is still high. The lower end of the inlet port communicates with the cylinder through a slot cut in the wall of the piston.

It will be noticed that a device to prevent firing-back is fitted

in the inlet port, consisting of a nest of thin metal plates. The object of this device is to split up the gases and expose them to a very large cooling-surface. In the event of flame passing back into the inlet port, this large surface is intended so to chill the burning gases that combustion cannot continue between the plates, and no flame can pass back and ignite the charge in the crankcase. The author is, however, very doubtful whether such a device can be effective, owing to the high velocity of the gases and, consequently, the extremely short time during which they are exposed

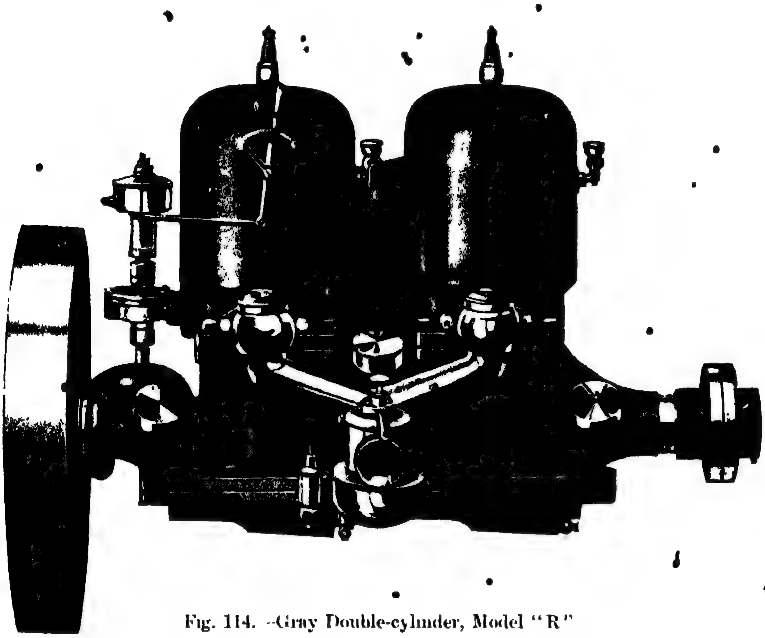


Fig. 114.—Gray Double-cylinder, Model "R"

to the cooling-surface. Back-firing into the crankcase is one of the greatest evils of engines of this type, and both the author and many others have experimented with similar devices, but without success.

In the Gray engine, shown in fig. 114, the cylinder, crankcase, and cylinder-head form three separate parts, but in the smaller sizes the cylinder and head are cast in one piece. The intake and exhaust pipes are both castings, and are held in place by means of steel clamps and dowel-pins, a very simple and neat arrangement. The exhaust pipe in all the larger sizes is water-jacketed for a short distance from the cylinder, and it is usual in marine engines to admit a portion of the circulating water into the exhaust pipe, where it mixes with and cools the exhaust gases, thus both helping to silence the engine and to keep the exhaust pipe cool. The sizes

made by the Gray Motor Company range from 3 to 12 horse-power per cylinder, the dimensions of the smaller size being $3\frac{1}{2}$-in. bore and $3\frac{1}{2}$-in. stroke, and the normal speed 600 R.P.M. The largest cylinder has a bore of $5\frac{3}{4}$ in. and a stroke of 5 in., the swept volumes being 33·5 and 130 cu. in. respectively.

Tests carried out at the Purdue University in 1909, on a two-cylinder 12-horse-power engine similar to the one illustrated in fig. 114, and having cylinders $4\frac{3}{4}$-in. bore and 4-in. stroke, gave the following results.

The specific gravity and calorific value of the fuel are not stated; but, since petrol was used, it will be fairly correct to take these as specific gravity 0·725, calorific value 18,500 B.T.U.s per pound lower value, and the brake thermal efficiency is calculated on that basis. The mechanical efficiency, indicated horse-power, and fluid losses are not recorded.

Speed (R.P.M.).	B.H.P.	Fuel (lbs. per B.H.P. hour).	ηp (Brake Mean Pressure)	Brake Thermal Efficiency	Specific Power = cu. in. swept per Stroke per B.H.P.
				Per cent	
909	14·11	1·18	44	11·6	
950	14·32	1·17	42·2	11·7	
996	14·59	1·10	41·0	12·4	
1010	14·75	1·03	39·5	13·3	
1071	14·72	1·11	38·5	12·3	9·65
1084	14·39	1·07	36·8	12·8	
1111	14·25	1·07	35·7	12·9	
1122	13·89	1·17	34·6	11·7	

Comparing these results with those which Professor Watson obtained from the Day engine, it will be seen that they are inferior; but the author, judging by his own experience with engines of this type, considers that Professor Watson's results are very much better than are usually obtained. The specific power of the Gray engine is 9·65 cu. in. swept volume per maximum horse-power; that of the Day engine 7·3 cu. in. before the alteration to the ports, and 5·6 cu. in. after. The maximum ηp value of the Day engine is 53 lb. per square inch at 922 R.P.M., and that of the Gray 44 lb. per square inch at 909 R.P.M., the piston speed being very nearly the same in both cases. But since in the Gray engine the ηp value increases uniformly as the speed decreases, it is probable that the maximum value is somewhat higher at a lower speed. The best brake thermal efficiency of the Day engine was 16·6 per cent before the alteration to the ports, while the best brake thermal efficiency

of the Gray engine is only 13·3 per cent, a somewhat poor result. It is probable that the considerable difference in the results obtained from these two apparently similar engines is to be found in the exact port timing, and possibly in the carburation. Professor Watson employed a carburettor with a hand-controlled needle adjustment for the petrol jet; with such an arrangement he was probably able to obtain the best possible conditions, so far as carburation was concerned; but this method is of course only applicable for a laboratory test. It would be useless under the practical conditions of everyday running, since it necessitates hand adjustment of the petrol supply for every change of load or speed.

The Petter Engine.—Messrs. Petters, Limited, of Yeovil, have recently put on the market a small two-cycle engine of the crank-chamber displacement type, designed to run on paraffin, and to develop 5 brake horse-power. In this engine, air only is compressed in the base-chamber, and the fuel is taken up by the air during its passage from the crank-chamber to the cylinder. The engine is governed by throttling the air supply from the crankcase to the inlet ports. The control of the fuel is effected by the suction in the crankcase, which draws a certain proportion of the fuel out of the fuel chamber into a small nozzle, situated in the inlet ports. The effect of restricting the passage of air from the crankcase to the cylinder is to reduce the suction in the crankcase, and also the suction on the fuel. Hence less air is taken into the crankcase, and less fuel into the fuel nozzle, and the proportion between air and fuel more or less retained. No vaporizer is employed, and such vaporization as takes place occurs in the cylinder.

Owing to the high velocity of the entering air, the paraffin will be thoroughly atomized, and partially vaporized by the hot cylinder walls and piston, and still hotter exhaust gases in the cylinder. This method of carburetting paraffin is rather attractive, in that it is not necessary to pre-heat any of the air required for combustion. Consequently, both a greater weight of air can be admitted, and a higher compression employed. Provided the fuel is thoroughly atomized, and intimately mixed with the necessary air for combustion, it does not matter very much whether it is vaporized or not, for there will be very little time for precipitation during the short period between its admission and combustion.

In this respect it would seem that the two-cycle engine distinctly gains over the four-cycle, for if fuel were admitted to the cylinder of a four-cycle engine during the suction stroke in the same manner,

a very large proportion of it would precipitate upon the cylinder walls during this period, nor would it have the same chance of being vaporized, owing to the much smaller proportion of hot exhaust gases present in the cylinder. In other respects the Petter engine is very similar to the two engines previously described, but instead of using inlet ports to the crank-chamber, a large leather flap valve is employed, giving an ample area with very little inertia. The use of leather valves for this purpose is only practicable for engines in which air only is taken into the crankcase, for if petrol be taken in it is liable to destroy the leather. Care must also be taken to ensure that the temperature of the crankcase shall never be high enough to damage the leather.

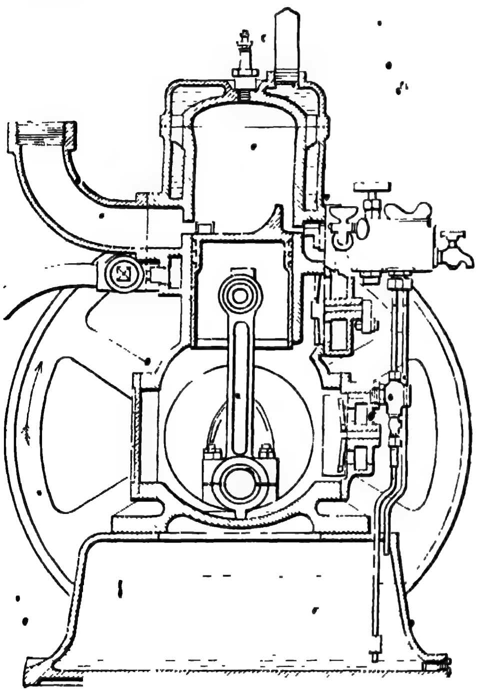


Fig. 115.—Petter "Junior" Stationary Engine

It will be observed from the illustrations (figs. 115 and 116) that in the Petter engine the cylinder and crank-chamber are cast in one piece, and a separate water-jacketed cylinder-head is fitted. This certainly makes for cheap construction, and would seem to be entirely satisfactory. Large inspection doors are provided on either side of the crank-chamber, through which the connecting-rod can be inspected, or, if necessary, disconnected, while to draw the piston the cylinder cover must be removed.

No particulars of any independent tests carried out on one of these engines are obtainable, but the makers state that the fuel consumption is approximately 4·1 lb. per hour. If the power be taken as 5 B.H.P. as stated, then the fuel consumption works out at 0·82 lb. per B.H.P. Taking the lower calorific value of the paraffin as 19,000 B.T.U.s per pound, the brake thermal efficiency becomes 16·3 per cent.

Three examples of crank-chamber scavenging engines have now been dealt with: the Day engine, using three ports and no valves;

the Petter engine, using an inlet valve and no port to the crankcase; and the Gray engine, using both an inlet valve and a port. These form the three leading types, and, although there are an immense number of similar engines on the market, they all differ from those described above only in the most trifling details. The type of two-cycle engine in which the fuel is injected into the cylinder at the end of the compression stroke will be considered at a later stage.

The Out-board Motor.—Yet another adaptation of this type of engine is for what are known as "out-board motors", that is to

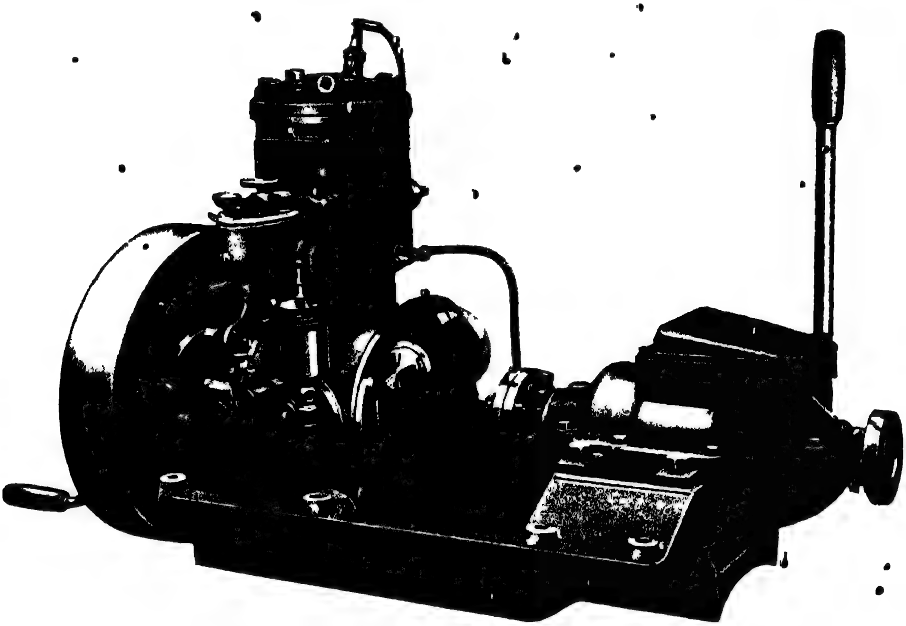


Fig. 116.—Petter "Junior" Marine Engine

say, small self-contained units consisting of an engine driving a propeller, which can be clamped to the stern of a dinghy or rowing-boat, and so convert it into a motor-boat. These little appliances, which originated in America, are rapidly becoming popular, for they can be very easily transported, and can be attached to practically any type of boat in a few minutes. A typical example of one of these motors, known as the Evinrude, is shown in fig. 117. It consists of a small three-port two-cycle engine of from $1\frac{1}{2}$ to 3 horse-power, mounted horizontally, with one side of the crank-shaft extended downwards, and running in a long brass sleeve. At the end of this sleeve is a small, torpedo-shaped box, carrying the horizontal propeller-shaft and propeller, which is driven from the extended crankshaft by bevel gearing.

A simple reciprocating plunger-pump for the circulating water is combined with this gear-box, and driven by means of an eccentric

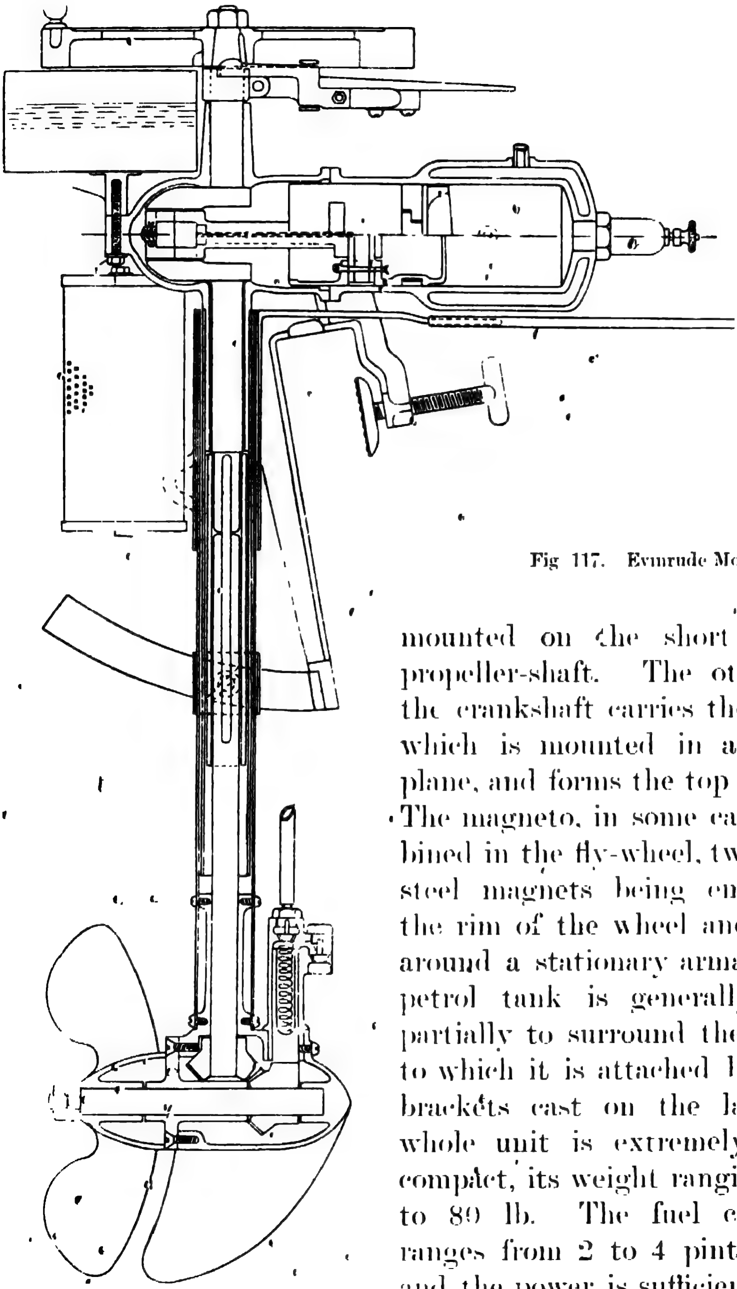


Fig 117. Evinrude Motor

mounted on the short horizontal propeller-shaft. The other end of the crankshaft carries the fly-wheel, which is mounted in a horizontal plane, and forms the top of the unit. The magneto, in some cases, is combined in the fly-wheel, two hardened steel magnets being embedded in the rim of the wheel and revolving around a stationary armature. The petrol tank is generally arranged partially to surround the crankcase, to which it is attached by means of brackets cast on the latter. The whole unit is extremely neat and compact, its weight ranging from 50 to 80 lb. The fuel consumption ranges from 2 to 4 pints per hour, and the power is sufficient to propel an ordinary sea-going dinghy at 3 or 4 knots.

The principal objection to these little out-board motors is the

serious vibration that they cause, owing to the high speed at which the engine runs, and the lack of rotary balance. To obviate this, a twin-cylinder opposed engine, known as the Archimedes, has been put on the market by a Swedish firm, and has two horizontal opposed cylinders. The pistons are connected to two cranks set at 180 degrees, so that the two pistons approach and recede simultaneously. Combustible mixture is drawn into the crankcase through a third port, and is compressed between the two pistons until the inlet ports are uncovered, when it enters the two cylinders. It is then compressed and ignited in the two cylinders simultaneously. By this means a perfect rotative balance is obtained, but the turning moment, and therefore the reactionary balance, is the same as though a single-cylinder engine of equal total capacity were employed. Owing, however, to the extremely high speed at which these engines run, the reactionary balance is a matter of very little consequence.

In practice these little engines run very smoothly, and, thanks to the perfect rotational balance, it is possible to run them at a much higher speed than the single-cylinder type, and therefore at the maximum of efficiency or specific power. In the author's opinion, the expense of adding an extra cylinder is thoroughly justified, for not only does it practically eliminate all vibration, but also allows of greater power for a given weight. Since the propeller is driven by gearing in any case, its speed is independent of that of the engine, and any reduction, within reasonable limits, can be provided in the gear-box. The Archimedes out-board motor is made in two sizes, 2 horse-power and 5 horse-power, the latter size being particularly intended for small sailing-yachts. In this case the complete equipment can be detached when the yacht is under sail, and the usual objection to auxiliary motors, namely, the drag of the propeller, is therefore removed. The vibration caused by some of the single-cylindered out-board motors, besides being most objectionable to the occupants, may be dangerous to the structure of the boat itself, which, as a general rule, has not been built to withstand severe vibration.

The Bessemer Engine.—The Bessemer engine, shown in figs. 118 and 119, is a two-cycle gas-engine of a somewhat different type from any that have been described so far. In this engine, the front side of the power piston is employed as a scavenging piston, but instead of totally enclosing the crankcase, the front end of the cylinder is closed, and a piston-rod and cross-head employed. This

reduces the clearance losses, as compared with crankcase scavenging, and also permits of the free lubrication of the crankshaft and connecting-rod bearings; but it is open to the serious objection that, with reduction of the clearance, the compression of the charge before the inlet ports open is proportionately increased, and since the compressed gases are afterwards released into the cylinder, when the inlet ports are uncovered, the work done upon them represents a dead loss. Hence, not only are the fluid losses increased, but the high velocity with which the charge enters the cylinder, due to its high initial compression, increases the degree of diffusion between

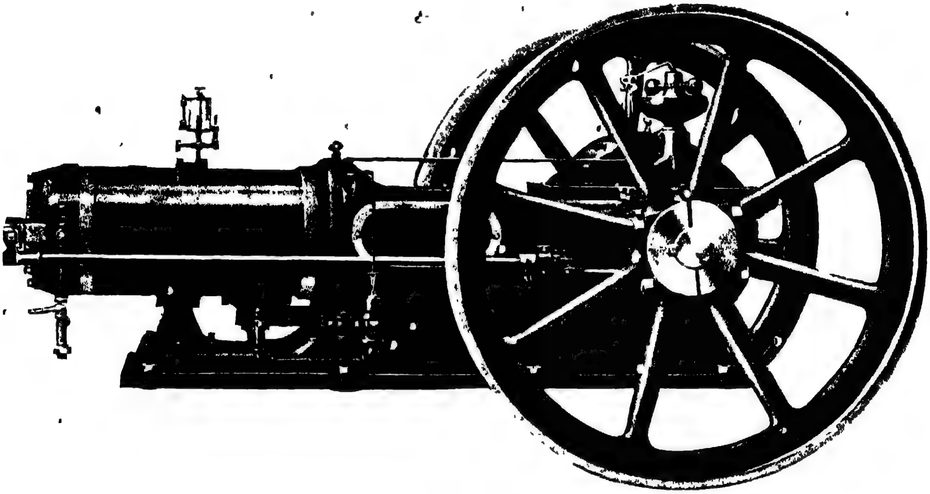


Fig 118.—Single-cylinder Bessemer Gas-engine

the incoming gases and the exhaust, and makes regular running on light loads impossible.

To illustrate this, let us suppose that the clearance in front of the piston, when the latter is at the end of the stroke, is just equal to the swept volume; then the gases will be compressed into half their initial volume, and their pressure raised to

$$(2^{1.4} \times 14.7) - 14.7 = 24 \text{ lb. per square inch,}$$

assuming adiabatic compression. The pressure in the cylinder, at the time of the opening of the inlet ports, will probably be in the neighbourhood of 3 lb. per square inch, and the gases will enter with a velocity corresponding to this pressure difference. The mean pressure on the front of the piston during the compression stroke will amount to about 10 lb. per square inch, plus about 1 lb. per square inch during the suction stroke, and, taking the mean pressure

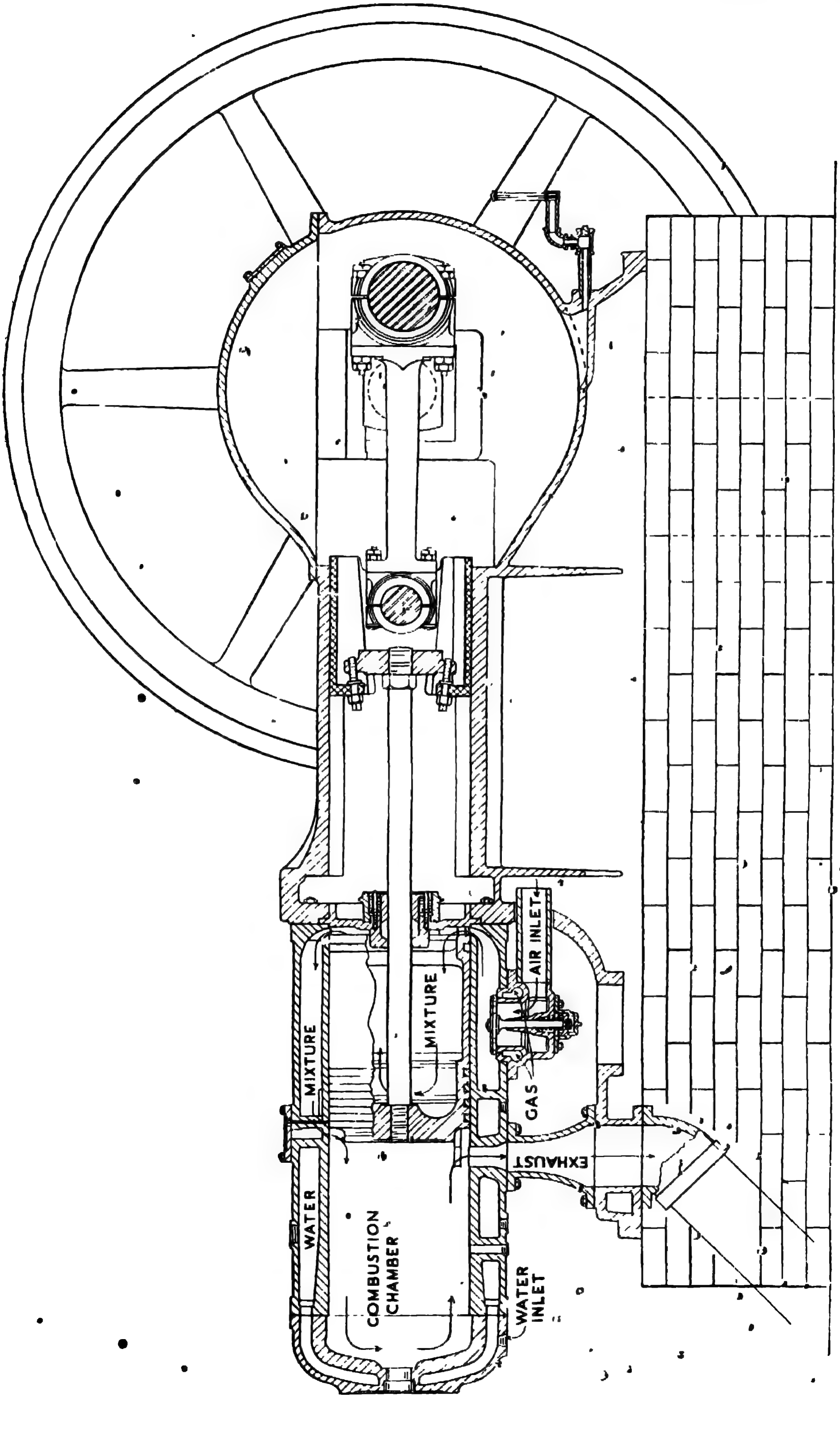


Fig. 119.—Bessemer Engine

of the expansion stroke at 60 lb. per square inch, it will be seen that the fluid losses on the delivery stroke amount to 18 per cent of the indicated horse-power. If the clearance is increased, the negative work will be diminished; but the proportion of charge that enters the cylinder will also be diminished, owing to the greater clearance losses, and the specific power of the engine reduced.

With this arrangement of scavenging it is necessary to arrive at some compromise between the low specific power of the engine on the one hand, and the high fluid losses on the other. In so far as fuel consumption and thermal efficiency are concerned, this engine is probably worse than the ordinary crankcase scavenging type, for the high velocity of the gases entering the cylinder will probably tend to increase the loss through the exhaust ports, and the fluid losses, as already shown, are greater. The indicator diagrams shown in fig. 120, were taken from a single-cylinder Bessemer engine of 16-in. bore and 20-in. stroke, developing 80 B.H.P. at 180 R.P.M., and running on natural gas, which in most of its characteristics is akin to petrol vapour. The full-load diagram is similar to that of the Day engine, and shows a mean effective pressure of 62 lb. per square inch, corresponding to an indicated horse-power of 112. The mechanical efficiency is, therefore, 71 per cent, and since the frictional losses in an engine of this size, using a separate cross-head and running at a piston speed of only 600 ft. per minute, will probably not exceed 6 lb. per square inch, the fluid losses become 19 per cent, which is about what one would expect.

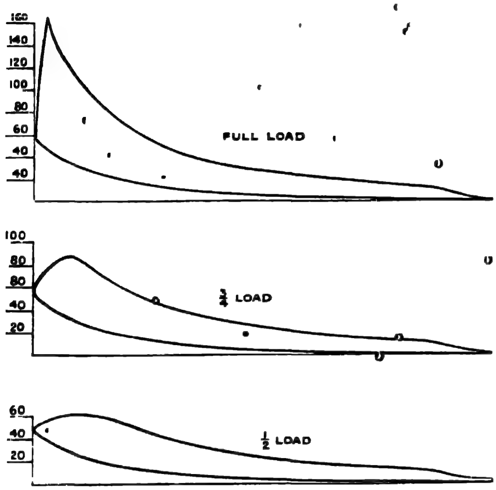


Fig. 120. - Indicator Diagrams. Bessemer Engine

From examination of the half-load diagram it will be seen that this bears all the evidence of retarded combustion, due probably to the complete diffusion of the fresh charge, and exhaust gases in the cylinder. It would seem that if the charge were still further reduced, combustion would either not take place at all, and the engine would "four-cycle", or would be so far delayed as to ignite the next charge on entry.

As showing the influence of stratification, it is interesting to compare this diagram with the half-load diagram taken from the Ricardo engine. If we examine the mechanical features it will be observed that, on the outward stroke, gas and air are drawn into the clearance space in front of the piston through the small spring-loaded poppet valve, which also serves as the mixing valve. The valve seating is flat and very wide, and has a ring of small holes drilled all round the face. Air enters through the centre of the seating, and gas through the small holes, the exact proportion being governed by the number and size of the latter. By this means the single admission valve serves also as the mixing and proportioning valve. The mixture so formed is very thorough, since the air has to pass over the seating at a high velocity, and in so doing meets a large number of jets of gas issuing from the small gas inlet holes. From the valve, the mixed gases enter the annular chamber surrounding the front portion of the cylinder, and also the front part of the cylinder itself, the purpose of the annular chamber being merely to increase the clearance space.

The head of the piston is made unusually thick in order to conduct the heat from the centre away to the cylinder walls as rapidly as possible; in this respect the use of a piston-rod very greatly assists in the withdrawal of heat from the centre portion of the piston. A deflector of a type somewhat similar to that used on the Day engine is provided on the head of the piston, to prevent short-circuiting through the exhaust ports. The cylinder is cast in one piece with the water-jacket, and no separate liner is employed. The cylinder-head is a plain dome-shaped casting of the simplest possible design, and there are no valves or valve ports to complicate the casting, or interfere with the free circulation of the cooling-water. The only passage through the cover is the small hole required for the ignition plug which is fitted in the centre. Such a combustion head as this is simple to cast, and should be very free from internal stresses due to casting, or to temperature differences in working, for the thickness of the inner wall can be almost uniform. The employment of an external and adjustable cross-head, in a position where it can be kept cool and freely lubricated, will make for higher mechanical efficiency, besides reducing the wear on the cylinder walls. This is an important consideration in engines as large as 80 horse-power per cylinder and employing no separate liner, though in this case it must be admitted that the cylinder is such a simple casting, and requires so

little machine work, that it is probably nearly as cheap to renew it entirely as to renew the liner of other engines.

A stuffing-gland is provided where the piston-rod passes through the front end of the cylinder. This gland is exposed only to the pressure in the front end, which is comparatively low, and is not subjected to any very high temperatures, so that little trouble need be anticipated from this source. The engine throughout is designed for rough work with very little skilled attention, and in cases where fuel is abundant and cheap it probably fulfils its purpose admirably. It is very extensively used in America, especially in the natural-gas and oil fields, for such purposes as pumping, air-compressing, and other contracting work. Its low first cost and extreme simplicity make it suitable for all purposes where power is required for temporary purposes.

The author has seen a very large number of these engines at work in the zinc and lead mines in Oklahoma and Missouri, U.S.A., using natural gas having a lower calorific value of about 950 B.T.U.s per cubic foot. They are also extensively used for pumping oil from the oil wells in America, using "casing head" gas, which consists largely of methane, and has a calorific value of from 1000 to 1100 B.T.U.s per cubic foot. For this purpose, a single-cylinder engine of from 30 to 40 horse-power is generally used, and the fuel consumption is said to average about 18 cu. ft. per B.H.P. hour, corresponding to a brake thermal efficiency of from 12·7 per cent to 14 per cent. The Company state that, during the last fifteen years, Bessemer gas- and oil-engines of an aggregate horse-power of over 400,000 have been supplied, in powers ranging from 8 horse-power to 80 horse-power per cylinder.

SUMMARY OF TWO-CYCLE ENGINE DESIGN

Before leaving the subject of two-cycle engines it would be well to sum up the main conditions which appear to be essential to the success of a two-cycle gas- or petrol-engine. In the first part of this volume the different methods of scavenging have been described, and later, practical examples of engines using each of these methods have been dealt with.

(1) The first condition is that the quantity of fuel lost through the exhaust ports shall be reduced to a minimum. To effect this, either primary and secondary air scavenging must be employed, or particular care must be taken to prevent diffusion between the

incoming charge and the exhaust gases. (2) In order to permit of regular running under varying loads, means must be provided whereby a small proportion of pure uncontaminated fresh charge is always present in the vicinity of the igniter. (3) In order to enable the engine to compete commercially, the cost and weight per horse-power must be less than that of a four-cycle engine, to balance the superior thermal efficiency of the latter. This means that high specific power must be obtained from a given size of cylinder. (4) The engine must be mechanically simple, and free from complicated castings, especially if the latter are exposed to the heat of combustion.

Stratification.—1. It is evident that whether air scavenging be employed or not, stratification should always be aimed at. If it be supposed that the incoming air passes down the cylinder in a perfectly stratified form, and that there is no admixture of, or diffusion with, the exhaust gases, then it is clear that one volume of air will expel and exactly replace a similar volume of exhaust gases, and that no air will pass out of the exhaust ports until the whole of the products of combustion are expelled. On the other hand, if the air is, as it enters, completely and instantaneously mixed with the exhaust gases, the formula $x = 1 - e^{-y}$ holds, where y is the charge admitted and x that retained. This, for one cylinder volume of charge gives 63·2 per cent of air retained in the cylinder. In practice, of course, neither of these conditions obtain, but the aim of the designer should always be to approach the first condition as nearly as possible.

To obtain good stratification it is necessary (1) that the entering charge shall be free from disturbing eddies, and shall pass down the cylinder at as low a velocity as possible. To effect this, the contour of the cylinder walls should be made to follow the natural stream lines of the entering fluid as nearly as possible. Further, to prevent the charge entering at a high velocity, it should not be previously compressed to a higher pressure than is absolutely necessary to overcome the exhaust back-pressure; that is to say, the relative movements of the pumping and power pistons should be such that the air is not delivered from the pump cylinder before, or at a greater rate than, it can enter the working cylinder, and receivers should not be employed.

(2) It is necessary that the release of the exhaust gases shall not be too sudden, or the gases remaining in the cylinder will be sub-

jected to violent disturbances and eddies. For this reason the exhaust ports should not be too large, and should be tapered, so that their opening is gradual. These conditions, while making for good stratification, do not make for rapid or complete combustion, for the gases will be in a state of stagnation during the compression stroke. To obviate this, and to produce violent turbulence in the gases at the time of ignition, the author favours the use of a small auxiliary igniting chamber. This may be charged with fresh gases, either by deflecting a portion of the main charge into it, or by charging it separately during the early part of the compression stroke. If the gas in this chamber be ignited, it will, owing to its rapid rise of pressure, penetrate the main body of the charge at a high velocity, producing violent turbulence in the gases, and igniting them at the same time.

It is evident that if perfect stratification were obtainable there would be no gain whatever in using separate air scavenging. It is because perfect stratification is apparently unattainable that air scavenging is employed. When scavenging with combustible mixture there is always a certain loss of unburnt gas through the exhaust, but with careful attention to the conditions governing stratification this can be reduced to a reasonably small proportion, provided that the quantity of mixture admitted is not too great. The author has found that, with careful design, the loss of unburnt gas is almost negligible up to indicated mean pressures of about 70 to 80 lb. per square inch when using petrol as fuel. Above this, however, it begins to be apparent, and at 100 lb. per square inch the loss is serious. For high mean pressures, therefore, it would probably be best to use air scavenging, since by that means the weight of air that can be retained in the cylinder without loss of fuel is greater. But it must be remembered that the use of air scavenging will increase the frictional and fluid losses to a very large extent, and the mechanical efficiency, especially at the lighter loads, will be reduced, so that the brake thermal efficiency may be no higher. In the case of small engines the author has found that separate air scavenging actually lowers the brake thermal efficiency, the difference being very slight at full load, but very marked indeed at or below half load.

Condition 2, regular running on light loads, can be met by good stratification, and so placing the igniter that it is always surrounded by comparatively pure combustible mixture. This appears at first sight to be simple, but is not easily accomplished; in practice

it is probably best to place the igniter in a separate chamber, as described above.

Condition 3, high specific power, can only be met by careful attention to scavenging, so that the greatest possible weight of air can be forced into the cylinder, with the minimum of both friction and fluid losses, and of mechanical complications.

Condition 4, mechanical simplicity, is probably best met by using the inverted U type of cylinder. Engines like the Korting involve the use of complicated cylinder covers of considerable thickness and unsymmetrical shape; such covers are liable to fracture owing to irregular expansion.

In constant-pressure two-cycle engines, such as the Diesel or Semi-Diesel, the problem of scavenging is very much simplified, since the fuel is not admitted until the completion of the compression stroke, and the control both of speed and power is carried out by varying the quantity of liquid fuel admitted, without any alteration to the scavenging. In these engines, all that is required is that the cylinder shall, at each revolution, be charged with as great a weight of pure air as possible, and that this shall be done with the minimum expenditure of power. The quantity of air taken in is at all times the same per stroke, and is generally independent of the power or speed of the engine, though it would be desirable on the score of mechanical efficiency to reduce the quantity of air on light loads. All that has been said with regard to stratification and diffusion applies to these engines also, though in a lesser degree, while on the other hand turbulence is of even greater importance.

There certainly seems to be a future before the two-cycle engine, if means can be found to reduce the fluid and friction losses and increase the specific power. In large plants, especially of blowing-engines, the air-scavenging pumps could be abolished, and their expense and friction saved, by scavenging the cylinders with air from the main delivery pipes of the blowing-tubs, or from a single large turbo-blower. This could serve all the engines in one power-house, and its efficiency would be very much higher and first cost lower than if each engine were fitted with separate pumps, but in this case some means would have to be found for distributing the air equally to all the cylinders. Such a blower, combined with a steam turbine, might easily be driven by low-pressure steam, generated from exhaust boilers. The heat otherwise lost to the exhaust might thus be utilized to perform the whole of the air scavenging,

and so effect a saving of fluid and friction losses amounting to some 7 or 8 per cent of the total indicated horse-power.

The specific power can best be increased by increasing the weight of air present in the cylinder when compression commences. This may be accomplished either by cooling the air before it enters the cylinder, and so increasing the weight of a given volume, or by throttling the exhaust, and so raising the pressures throughout the whole system. Experiments carried out by Professor Junkers on a large Oechelhauser engine at Hoerde, Westphalia, in which an inter-cooler was fitted between the pump cylinder and the power cylinder, gave very satisfactory results. By cooling the scavenging air from 192° F. down to 77° F., the output of the engine was increased from 390 to 460 horse-power, an increase of nearly 20 per cent in the specific power. The increase of weight of a given volume of air due to this difference of temperature would be in the proportion of 683 to 568, or slightly over 20 per cent, so that the specific power varied directly in proportion to the weight of air admitted; which is as might be expected if all other conditions remained the same. It is reported that this increase of power was obtained without any increase in the work of the pump, or in the total amount of heat carried away by the cooling water. That the total quantity of heat carried away by the jackets was not increased is quite possible, for, although the quantity of heat was increased by 20 per cent, the temperatures throughout the whole cycle were reduced. The second method of increasing the specific power, namely by throttling the exhaust, has also been the subject of a great many experiments carried out by Professor Junkers. It is evident that if the exhaust be so throttled that the back-pressure amounts to, say, 1 atmosphere, and if the same volume of air be present in the cylinder, the pressures throughout the whole cycle will be doubled without any increase in temperature, and consequently the specific power of the engine will be doubled, while the proportionate heat loss and mechanical friction will be reduced. On the other hand, the work on the pump will be greatly increased, owing to the greater back-pressure. The net result is that the indicated thermal efficiency remains much about the same, but, owing to the greater proportionate work on the pump, the brake thermal efficiency is lower than when working under normal conditions.

The indicator diagrams illustrated in fig. 121, and shown super-imposed, were taken, the smaller at normal full load, and the larger when the exhaust was so far throttled as to raise the pressures by

about 50 per cent. In this latter diagram the mean effective pressure is no less than 220 lb. per square inch, a truly remarkable result. This system of supercharging, combined with inter-cooling, seems

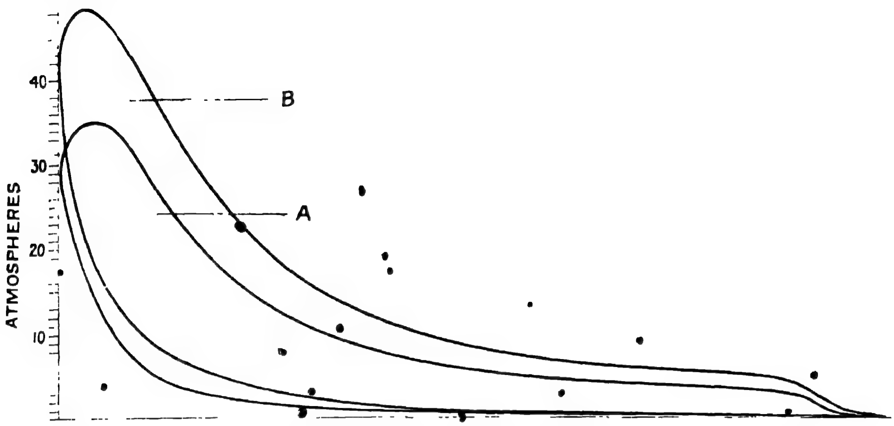


Fig 121 - Diagrams Superimposed showing Supercompression Effect Junkers

A, Normal engine diagram. B, Supercompression engine diagram.

to be the most promising method of increasing the specific power of two-cycle engines, but the same result can be accomplished in the case of four-cycle engines even more advantageously.

CHAPTER XXI

SOME TYPICAL HORIZONTAL ENGINES

Although the ordinary gas-engine was the earliest, and is still the best-known form of internal-combustion engine, yet it is not proposed to give up a great deal of space to the consideration of this type, partly because it has already been dealt with at considerable length, and partly because it is rapidly losing its predominant position, owing to the severe competition from electricity in the smaller powers, and from steam and Diesel and other oil-engines in the larger. The gas-engines in use at the present time may be divided into three classes:—

1. Small engines up to about 30 horse-power, used for general utility purposes, and generally supplied with town or illuminating gas.

2. Medium-sized engines of from 20 to 500 horse-power, generally used for shop-driving and for generating electricity in small outlying plants.

3. Large engines of from 500 to 5000 horse-power, using waste gases such as blast-furnace or coke-oven gas, and employed either for generating electricity or blowing blast-furnaces.

Small engines of the first class are used principally for general shop-driving, and have to compete with electricity for this purpose. Electricity can generally be purchased for small-power purposes at, approximately, 1*d.* per unit, which is equal to about 0·88*d.* per horse-power hour. A small gas-engine below 30 horse-power will consume on the average about 12,000 B.T.U.s per horse-power hour, with varying loads, which is equivalent to about 20 cu. ft. of average illuminating gas. With gas at 2*s.* 6*d.* per 1000 cu. ft., therefore, the cost of fuel amounts to 0·6*d.* per horse-power hour, but the electric motor has the advantage that its initial cost is lower, it occupies less space, and the cost of upkeep also is lower. Moreover, since it is much easier to start, and generally very much handier, the stand-by losses are less. Taking all these

points into consideration, the electric motor is generally to be preferred, and, with the rapid spread of cheap electricity throughout the country, the scope of the small gas-engine using illuminating gas is being narrowed down. It may be that the gas companies will see fit to supply gas suitable for power purposes at a much lower rate, as they probably could do; but, unless this happens, the small gas-engine is in danger of being driven off the market by its competitor, the electric motor.

In addition to the electric motor, it has to compete with the semi-Diesel type of oil-engine, using residual oil. Such engines, in normal operation and on varying loads, consume about 10,000 B.T.U.s per brake horse-power per hour. With residual oil at, say, 70*s.* per ton, the cost of fuel works out at only 0·2*d.* per horse-power hour, but such engines require heating up at starting, and are, generally speaking, less handy than gas-engines. Their influence has not, as yet, been very severely felt; but there is little doubt that when this type of engine becomes better known and understood, there will be but a limited scope for the ordinary gas-engine using illuminating gas.

The larger class of gas-engine, employing producer-gas made from coke, or anthracite, bituminous coal, wood refuse, &c., forms the cheapest source of power in existence. With coal at 16*s.* per ton, the cost of power is only 0·10*d.* per horse-power hour, based on a consumption of 1·17 lb. of coal per B.H.P. hour, which is a fair allowance. On the score of fuel economy, the producer-gas engine has nothing to fear from either the steam-, the Diesel engine, or the electric motor.

Unfortunately, however, gas-producers are in themselves a source of weakness. Both the producer itself and the gas-cleaning apparatus require very careful supervision, if successful results are to be obtained from cheap bituminous coal. Compared with an average steam plant of equal capacity and medium power, the gas-engine using producer-gas consumes less than one-third the quantity of fuel, but it lacks the flexibility and reserve capacity of the steam plant. Compared with electricity, the cost is about one-sixth, and compared with the Diesel engine it is half. It would seem that there is a considerable future before the gas-engine and producer-plant, not only for medium but also for comparatively large powers, if only the troubles which at present are experienced with bituminous coal producers can be eliminated.

In large producer-gas plants of over 1000 horse-power it pays

to instal ammonia recovery plant for the production of sulphate of ammonia, and since this by-product has a very high, though fluctuating, market value, the cost of generating power can still further be reduced.

The larger types of gas-engines using waste gases have been very extensively developed on the Continent, and especially in Germany. They are employed either for generating electricity or for blowing blast-furnaces, and their use is generally restricted to iron and steel works and coke-oven plants; in all of which cases waste gases are plentiful. They have, of course, to face a severe competition from steam, for steam is now very efficient for high-power outputs, and its efficiency has been enormously increased lately by the introduction of exhaust steam-turbines, uniflow engines, and other modern developments. The steam-engine, moreover, has certain marked advantages which give it a very substantial superiority over the gas-engine, the chief of which is its capacity for dealing with temporary overloads, and its flexibility generally.

Unless the large gas-engine can show a more marked superiority over steam than it is able to do at present, it is open to doubt whether this type of prime mover will survive. There are indications, however, that both the overload capacity and the efficiency of large gas-engines may be improved by the application of supercharging, and the more extensive use which is being made of the heat of the exhaust gases for steam-raising, &c.

The theoretical considerations influencing the design of gas-engines have already been discussed in the first volume, and it remains only to investigate a few typical examples of each of the leading types of engine now on the market, and to devote a certain amount of consideration to the question of governing and speed-control.

The Crossley Engine.—With regard to the smaller types of gas-engines using town gas, these have settled down to what may be described as a standard design, and there is very little difference between the engines made by the various manufacturers in England and abroad. The Crossley engine shown in figs. 122 and 123 may be regarded as typical of this class. The mechanical features call for very little comment. The cylinder-jackets and bedplate are cast in one piece, forming a very rigid construction. By this means the bearings for the side- or camshaft can be mounted upon the side of the bedplate, leaving the breech-end free. This is a point of considerable practical importance, for it

is necessary to remove and clean out the breech-ends from time to time, and this operation is a very tiresome one when the cam-

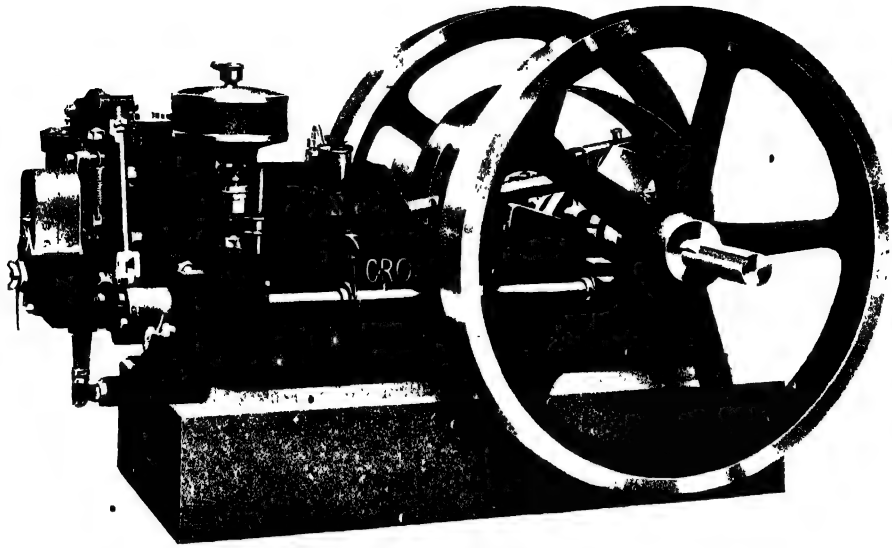


Fig. 122.—New Type of Small Gas-engine (Crossley)

shaft is carried in bearings attached to it. Practically all first-class gas-engines are now so designed that the breech-end can be removed without disturbing the camshaft or any other gear, except the pipe-connections. The cylinder-liner is bolted to the breech-end or combustion chamber, and passes through a cylindrical hole bored out at the front end of the water-jacket. The liner at this point is a free sliding fit in the jacket, and leakage of water is prevented by means of one or more rubber rings fitted into grooves turned in the liner, and acting practically as piston-rings. This form of joint is now almost universally employed, and is extremely satisfactory. It allows the liner to expand freely, and is perfectly watertight.

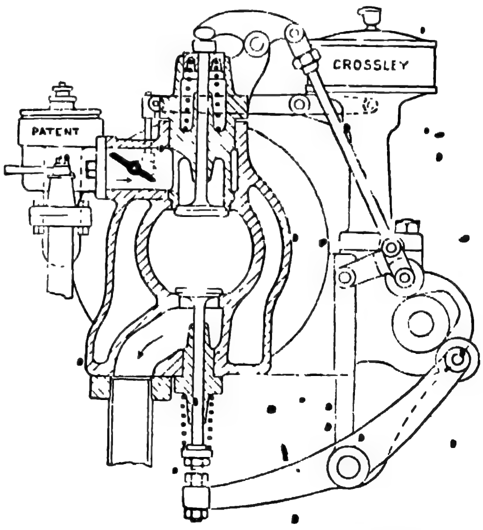


Fig. 123.—Section through Breech-end. Crossley Gas-engine.

The crankshaft is mounted on bearings, the lower half of which form an integral part of the bedplate. The bearing-surfaces are usually of white metal, but in the small sizes a special mixture of phosphor-bronze is frequently employed. Ring lubrication is relied upon, and it would be difficult to suggest any improvement upon this system, which is admirably simple and efficient in cases where the bearing-loads and speeds are not too severe. The connecting-rod big-end bearings are lubricated by means of wick-lubricators in the very smallest engines, and by means of centrifugal- or "banjo"-lubricators in the larger sizes. These centrifugal-lubricators are fed by a separate sight-feed lubricator, which can, of course, be replenished while the engine is running. The pistons are of cast iron, and are very long and heavy. Even at the low speed at which these engines run, they must have a prejudicial effect upon the mechanical efficiency. They are very carefully fitted to the liners, and the clearance allowed below the rings is extremely small.

The pistons and cylinder-walls are lubricated by means of an oil-pump, operated from a cam on the side-shaft, and the gudgeon-pin receives its supply of oil from the same source, suitable ducts being provided to lead a portion of the oil scraped from the cylinder-walls to the gudgeon-pin. The gudgeon-pin bearings are of phosphor-bronze, made in two halves, and are adjustable like the crank-pin bearings. The author considers that this is unnecessary in the smaller sizes of engines. The provision of adjustable bearings involves a considerable increase in the cost, and, worse still, in the reciprocating weights, as compared with a plain bronze lining or bush. It is necessary, in any case, to withdraw the connecting-rod and piston in order to make any adjustment, and in these circumstances it is quite as easy to replace a worn bush with a new one, as it is to adjust a split bearing. The gudgeon-pins are of mild steel, case-hardened, and ground to size, and under normal circumstances they show very little tendency to wear, but such wear as does occur is very local, owing to the small angle through which the connecting-rods oscillate. Hence, it cannot be taken up by any adjustment of the bearings, but the pin must be either re-ground or replaced.

The breech-end is a single casting, complete with its water-jacket; and though in the larger sizes this piece is always a source of weakness, in engines up to 30 horse-power per cylinder it does not present the least difficulty. Two valves only are employed, one for the admission of both gas and air which have previously been mixed, and

one for the exhaust. Both valves are vertical, and the inlet is provided with a detachable seating so that it can be easily removed. At the same time, the exhaust valve can be withdrawn through the opening left by the inlet-valve seating. The inlet valve is slightly larger than the exhaust, as indeed it should be. Both valves are mechanically operated, through the medium of rocking levers from the cams on the horizontal side-shaft. The side-shaft runs alongside and slightly below the centre-line of the cylinders. It is carried in ring-oiled bearings mounted on the side of the bedplate, and is driven from the crankshaft by means of spiral gearing, with a speed reduction of 2 : 1. Besides the cams for operating the valves, the side-shaft also drives the governor, the magneto for ignition, and the lubricator for the piston. In the smallest sizes of Crossley engine, governing is effected by hit and miss, in which case a separate gas valve is, of course, used operated by means of a separate cam and rocking lever. The end of the rocking lever is provided with a chisel edge, which engages with the gas valve through the medium of a sliding block, between the end of the valve stem and the "pecker". So long as the block is in its normal position, the chisel edge of the "pecker" engages a groove in the block, and so opens the valve; immediately the speed is increased above the normal, the governor raises the block out of reach of the "pecker", and the valve remains closed.

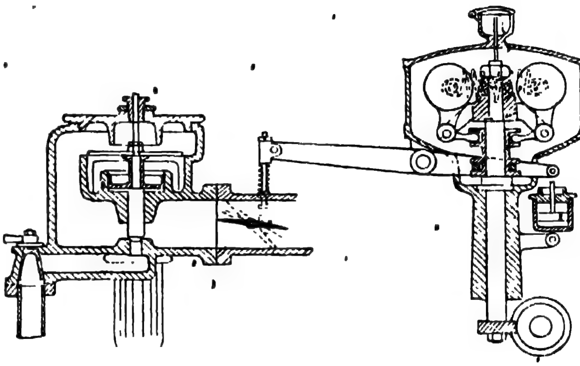


Fig. 124.—Crossley Gas-engine. Governor Gear.

In all the larger sizes of Crossley engines, from about 6 horse-power upwards, quantitative governing is employed. In sizes ranging from 7 to about 30 horse-power an automatic mixing valve is used, and the governor acts upon a butterfly throttle valve, which regulates the supply of mixture delivered to the cylinders. The type of mixing valve employed is illustrated in fig. 124, which also shows the governor and throttle valve. Air enters through an air-silencer, and passes thence by a passage (not shown in the diagram) to the space below the air-disk. This disk is mounted on a spindle, which also carries the gas-admission valve and a small

damper-piston to prevent it from fluttering. The suction of the engine lifts the disk, and with it the gas valve, so that both gas and air enter the throttle chamber. It will be noticed that the air is drawn past the gas-admission port at a high velocity, in order that it shall thoroughly mix with the gas. The proportion of gas to air is regulated by the proportional diameters of the air-disk and gas valve. The whole arrangement is very simple, and has the advantage that, should the engine stop from any accidental cause, the gas valve will immediately and automatically close, and thus any escape of gas will be prevented.

Another feature worth noticing is that the air has considerably farther to travel than the gas. This is important, because, when the main inlet valve of the engine closes, there is always a momentary reversal in the direction of flow of the fluid in the induction pipe, and a certain proportion will blow back and escape into the atmosphere before the mixing valve has time to close. By making the air-passages longer than the gas, whatever is lost in this manner will be air, and not gas. For running on petrol, the same mixing valve is employed, the only modification being that the gas valve is replaced by a tapered needle, which regulates the supply of petrol from a small jet.

This engine may be taken as representative of the smaller types of gas-engines as made by nearly all makers in this country and abroad. Such variety as exists is to be found only in the governor gearing, and in quite secondary mechanical details. The whole design of gas-engines of this class has become practically standardized. Qualitative governing is never employed on these small engines, and the choice lies between quantitative or "hit and miss"; but the latter, on account of the irregular turning movement, is now rapidly dying out.

The following test figures have been obtained from a small Crossley gas-engine of 25 B.H.P., when running at a speed of 260 R.P.M., and using town gas having a lower calorific value of 540 B.T.U.s per cubic foot:—

B.H.P.	Gas Consumption (cu. ft. per B.H.P. hour).	Brake Thermal Efficiency.
30	16·7	28·3 per cent
14·5	17·25	27·1 ,,
13·3	18·1	26·1 ,,
12·25	19·7	24·0 ,,
6·12	27·8	17·0 ,,

If the mechanical efficiency be taken as 88 per cent on full load, as it may well have been in this engine—which has large valves and comparatively light reciprocating parts—the figures become:—

B.H.P.	I.H.P.	Mechanical Efficiency.	Indicated Thermal Efficiency.
30	34·1	88 per cent.	32·1 per cent.
24·5	23·6	85·6 ,,	32·0 ,,
18·3	22·1	81·7 ,,	32·0 ,,
12·25	16·35	75 ,,	32·0 ,,
6·12	10·22	60 ,,	28·4 ,,

"Gardner" Engines.—The engine illustrated in figs. 125 and 126 is made by Messrs. L. Gardner & Sons, of Patricroft, Manchester, and resembles the Crossley engine in all but a few small details. The leading dimensions of this engine are as follows:—

Bore	8·5 in.
Stroke	16 in.
Number of cylinders	1.
Swept volume	912 cu. in.
Area of piston	57 cu. in
Compression ratio	6·26 : 1.
Maximum B.H.P.	24·4.
R.P.M.	240.
Piston speed	640 ft. per minute.
Brake mean pressure	88 lb. per square inch.
Diameter of valve ports	2·5 in
Lift of valves	0·625 in.
Effective area of opening	4·9 sq. in.
Ratio (piston area to effective area of valves) ...	11·6 : 1.
Weight of piston	47 lb.
Weight of reciprocating parts per square inch of piston area	1·33 lb.
Diameter of crankshaft	4 in.
Width of bearings	7·625 in.
Weight of connecting-rod	87 lb.
Weight of reciprocating parts	76 lb.
Projected area of main bearings	30·5 sq. in.
Length of connecting-rod between centres ...	40 in.
Ratio (length of connecting-rod to crank-throw $\frac{l}{r}$)	5·1.
Fuel consumption (cubic feet per B.H.P. hour full load)	16·4.
Lower calorific value of gas (B.T.U.s per cubic foot)	545.
Brake thermal efficiency	28·5 per cent.

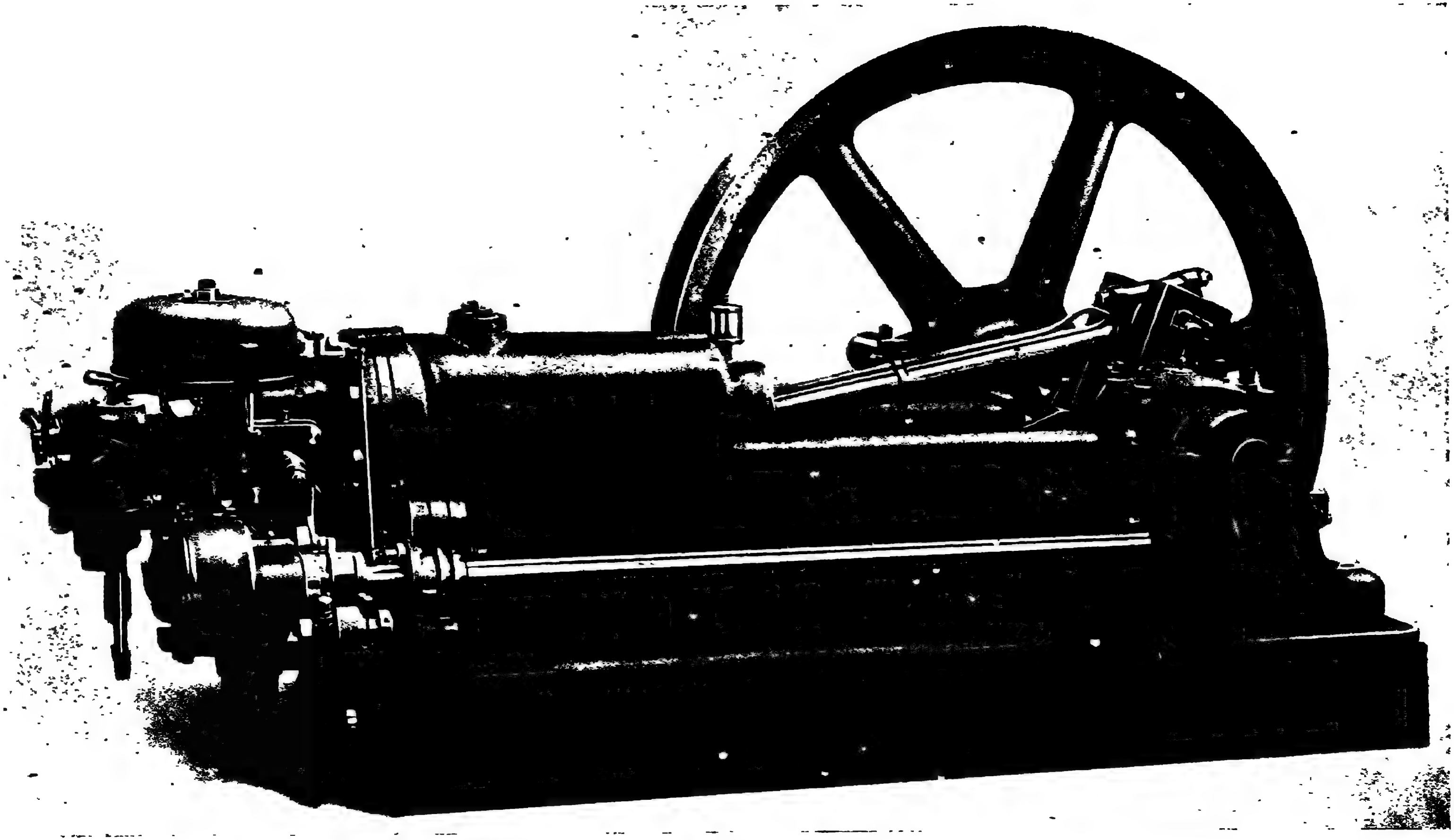


Fig. 125.—"Gardner" Engine

The particular engine to which the above figures relate was designed to run with producer-gas, hence the high compression ratio. Tests were carried out at Messrs. Gardner's works, using ordinary town gas as fuel. This was perfectly satisfactory so long as the engine was new, and the pistons and combustion chamber clean and free from carbon deposit, but it is probable that after long periods of running on town gas, with this high compression ratio, pre-ignition would be set up. Tests were carried out at "full", "three-quarters", "half", and "no load", and yielded the following results:

Load (B.H.P.).	R.P.M.	Mean Pressure (lb. per sq. in.).	Compression Pressure.	Gas per Hour (cu. ft.).	Gas per B.H.P. Hour.	Brake Thermal Efficiency.
						Per cent
24·1	240	88	160	402	16·4	28·5
18·4	242	66	145	337	18·2	25·7
12·5	245	44	100	276	22·0	21·2
—	249	—	45	150	-	—

The mechanical efficiency on full load was reckoned to be 89 per cent.

As the load is reduced, the fluid losses will increase, owing to the enlarged suction loop due to throttle-governing; but the mechanical losses will be slightly reduced, so that the net loss may be assumed as practically constant at all loads. The actual results will, therefore, probably be approximately as follows:

Load (B.H.P.).	I.H.P.	M.E.P. (lb. per sq. in.).	Mechanical Efficiency	Indicated Thermal Efficiency
			Per cent	Per cent
24·1	27·5	99·2	89	32
18·4	21·5	77·5	85 5	30
12 5	15·6	56	80 0	26 6
	3·1	11·2	—	9·7

On full load, the indicated thermal efficiency is 32 per cent. With a compression ratio of 6·26 : 1, the air standard efficiency is, approximately, 52 per cent, and the relative efficiency 61·4 per cent. The mean effective pressure, 99·2 lb. per square inch, is high for a gas-engine, and involves a mixture density of nearly 80 B.T.U.s per cubic inch. For this mixture density, the ideal indicated efficiency, taking into account the increase of specific

heat at high temperatures, is 73 per cent of the air standard, or 38·1 per cent. The actual indicated thermal efficiency is approximately 84 per cent of the ideal, the difference of 16 per cent being due to loss of heat to the cylinder walls during combustion and expansion, and early opening of the exhaust valve.

On the whole, the results are remarkably good for a small engine of only $8\frac{1}{2}$-in. bore, and running at such a low piston speed as 640 ft. per minute. Had the piston speed been higher, the loss of heat to the cylinder walls would have been less, but the mechanical efficiency would have suffered. The piston and reciprocating parts in this engine are exceptionally light as compared with usual gas-engine practice, so that considerably higher piston and rotative speeds could be employed without any appreciable increase in the mechanical friction: but it would be necessary, in that case, to increase the size of the valves, otherwise the fluid losses and volumetric efficiency would suffer severely.

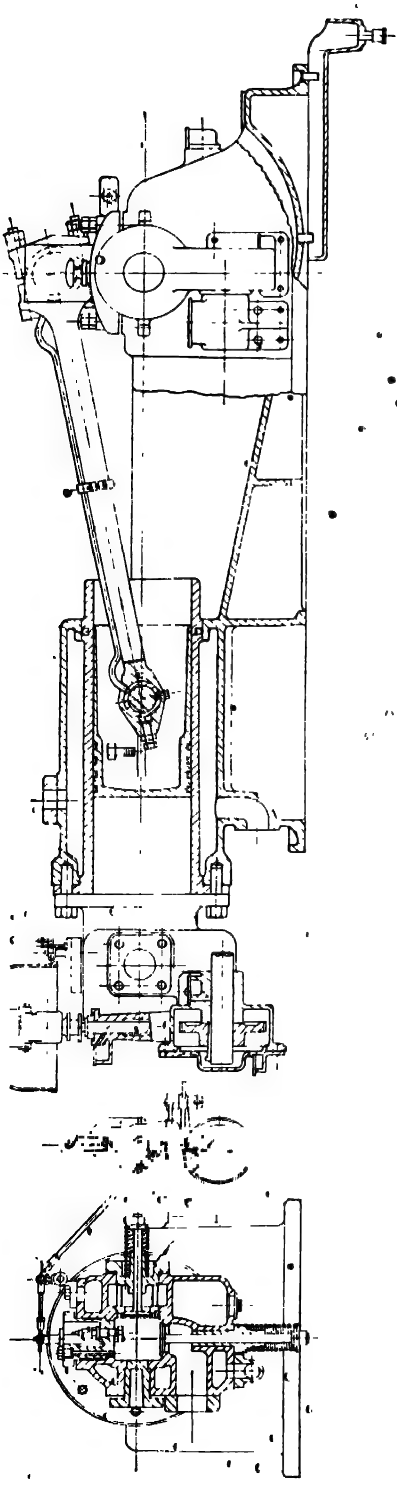


Fig. 126.—"H" Type Heavy "Gardner" Gas-engine

CHAPTER XXII

MEDIUM-POWER GAS-ENGINE

Engines for Shop-driving, &c.—Engines of the second class, for shop-driving, &c., are generally built in either the horizontal or vertical type, the choice depending upon—

1. The tools and the general habits of the manufacturers.
2. The purpose for which the engine will be used.

The arguments for and against the horizontal or vertical type are very numerous, and often complex, but as a general rule they may be reduced to one or other of the two reasons given above.

Where a high rotative speed is required, as, for instance, for generating electricity, forced lubrication and an enclosed crank-chamber become essential. Under these conditions the principal claim in favour of horizontal engines, namely accessibility, is removed, and the vertical type is to be preferred, because it occupies less floor-space and is somewhat easier to lubricate. Where, however, there is no object in high rotative speeds, and an open engine can be employed, the horizontal type has the advantage that—

1. Both the pistons and valves are more accessible.

2. For a given speed and power it is lighter, and, therefore, cheaper.

3. It allows of a very good form of combustion chamber and arrangement of valves, without introducing any difficulties in the way of valve operation. The same arrangement cannot be satisfactorily employed in the vertical type, because, in that case, the valves themselves would be horizontal, which is undesirable, except in engines of small size.

4. It allows the use of a longer stroke, with its attendant advantages.

For engines of from 15 to 150 horse-power per cylinder the fuel generally used is producer-gas, which is very much cheaper than town gas.

Producer-gas, however, has the following disadvantages:—

(*a*) The engine cannot be started instantly, because it takes an appreciable time to get the producer lighted up and under way.

(*b*) The composition of the gas is liable to vary between fairly wide limits, thus affecting both the mixture strength and governing.

(*c*) The calorific power of the gas is low, generally in the neighbourhood of 140 B.T.U.s per cubic foot, as against 550 to 600 B.T.U.s per cubic foot in the case of town gas. This means that the power output is necessarily reduced, because the quantity of gas required is larger and the air is, therefore, reduced. Hence, a lower mean pressure and lower mechanical efficiency. This, however, is largely offset by the fact that producer-gas permits of the use of a higher compression ratio, without risk of pre-ignition.

(*d*) Unless anthracite (or, at any rate good-quality and, therefore, expensive coal) be employed, trouble may arise from an accumulation of tar in the valves and governor gearing of the engine. This can be mitigated, but not entirely overcome, by the use of tar extractors.

It is not proposed, however, to deal with the extensive subject of gas-producers in this book.

The Tangye Engine.—Fig. 127 shows a Tangye engine of the type supplied to work in conjunction with the Tangye suction gas-producer, using anthracite coal. It is designed to give a continuous output of about 65 B.H.P. when running at a speed of 190 R.P.M. In general features it differs but little from the small Crossley or Gardner engines just described, the principal difference being in the governing, the latter being accomplished by varying the lift of the inlet valve.

On the inlet-valve stem is mounted a separate small valve which controls the admission of gas. This valve is fitted loosely on the stem of the main valve, and is held up to its seating by a light spring. A collar is provided on the main valve stem, slightly above the gas valve, in such a position that, as the main valve opens, the collar comes into contact with the gas valve, and opens that also. There is a small amount of clearance between the collar and gas valve, in order to give the air valve a slight lead during the first and last fractions of the valve travel, when air only is admitted, and the passages are thus cleared of any gas that may linger in them. With this arrangement, it is clear that, as the lift of the valve is increased, the quantity both of gas and air admitted to the cylinder

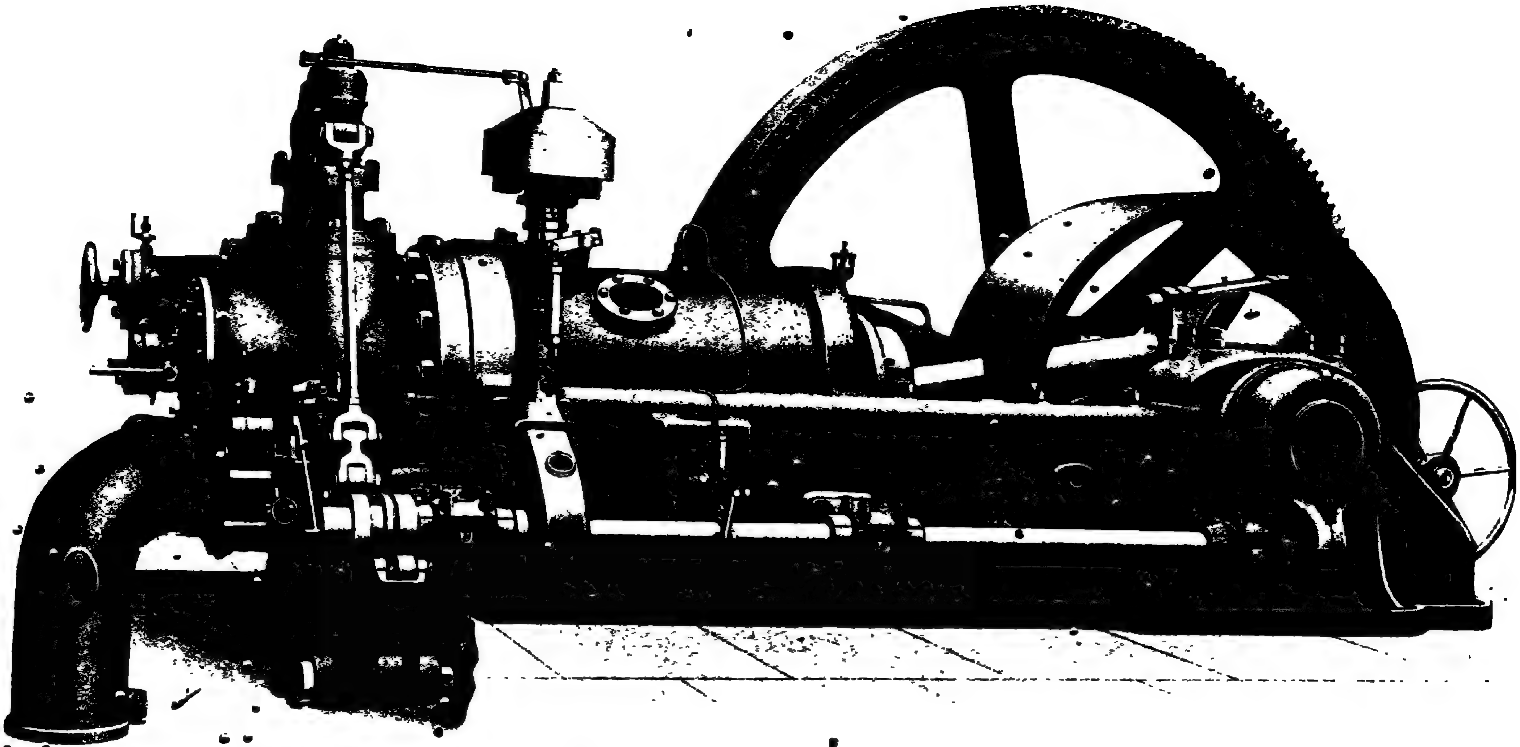


Fig. 127.—Single-cylinder Type Gas-engine with new Governor Gear

is correspondingly increased, while the proportion of each is adjusted by the relative diameters of the gas and air valve ports.

The method of operating and varying the lift of the valve is somewhat peculiar. A curved lever is pivoted at one end to the valve stem, and the other end is connected, through a push-rod and roller, to the cam mounted on the side-shaft. Above this curved lever is fitted another short lever of slightly greater curvature, one end of which is pivoted to a pin above, and slightly to one side of, the valve stem. The other end is connected to a screwed plug which can be raised or lowered by the partial rotation of a hollow cylindrical nut. The whole arrangement is shown in detail in fig. 128. As the movable end of the second short lever is lowered, the point of contact between it and the longer operating lever is brought farther away from the valve stem, and the lift of the valve is increased. The two curved levers have a rolling action, with the result that, as the valve lifts, so the point of contact, or fulcrum, shifts farther from the valve stem, and the velocity of opening increases. This arrangement provides for a gradual opening and closing of the valve, with rapid acceleration after it has once been started in motion. The lift of the valve is controlled by rotating the hollow cylindrical nut which is coupled to the governor.

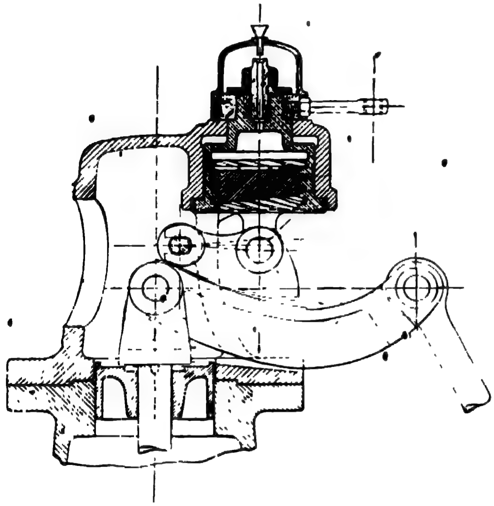


Fig. 128.—Tangye Governor Gear

The whole arrangement is ingenious, but it is evident that the "sensitiveness" of governing must depend upon the perfect freedom of movement of the screwed nut and plug. Any friction here, due to grit, &c., will throw a load on the governor, and will be liable to cause hunting. Moreover, the gear is not balanced, for there will always be an upward pressure upon the plug, which will tend to rotate the nut and so react upon the governor. To check this as far as possible, the nut itself is provided with a wide flange, which bears against the bracket supporting it, and whose friction prevents it from rotating.

The exhaust valve consists of a steel stem with a separate cast-iron head, which is very thick and heavy, in order to enable it to get rid of its heat readily, and to prevent distortion.

A separate renewable seating is provided in the combustion chamber for the exhaust valve, a provision which seems to be desirable in any but very small engines. In other respects the engine differs so little from those previously described that there is no need to devote any further space to it.

A series of tests were carried out on an 80 horse-power Tangye engine, with a suction producer using anthracite fuel, by Professor Mathot (of Brussels) in 1909. This engine had the following leading dimensions:—

Bore	$16\frac{1}{2}$ in.
Stroke	23 in.
R.P.M.	190.
Piston speed	728 ft. per minute.
Area of piston	214 cu. in.
Swept volume	4922 cu. in. = 2·848 cu. ft.
Compression ratio	6·3 : 1 (approx.).
Air standard efficiency	52 per cent (approx.)

[illegible]

	Test 1.	Test 2.	Test 3.
Duration	10 hours.	5 hours.	16 min.
R.P.M. (average) ...	190·86.	190·34.	189·25.
B.H.P. . .	68·68.	81·12.	88·76.
I.H.P.	81·56.	93·91.	—
Mean pressure ...	68·84 lb. per cu. in.	79·51 lb. per cu. in.	—
Mechanical efficiency	84·21 per cent.	86·35 per cent.	-
Compression pressure	152·89 lb. per cu. in.	159·29 lb. per cu. in.	
Maximum pressure ..	265·97 ,, ,,	356·99 ,, ,,	
Fuel consumption (lb. per B.H.P. hour) ..	0·720.	0·665.	—
Overall brake thermal efficiency	24·8 per cent.	26·9 per cent.	—

The brake thermal efficiency includes the losses in the producer. It is very doubtful whether the efficiency of the gas-producer could be higher than 85 per cent, and, in this case, the actual brake efficiencies of the engine, reckoned on the gas consumption, become 29·2 per cent at a load of 68·68 B.H.P. and 31·7 per cent at a load of 81·12 B.H.P. The mechanical efficiency is given by Professor Mathot in his report as 84·2 per cent in Test 1, and as 86·35 per cent in Test 2. If these figures be accepted, then the indicated

thermal efficiency becomes 34·7 and 36·7 per cent. These are exceptionally good figures for an engine running on producer-gas, and Professor Mathot remarks that they are the best he has ever attained.

The relative efficiency in the former case is 67 per cent, and in the latter 70·7 per cent, both these figures being based upon the somewhat doubtful assumption that the producer efficiency is 85 per cent in each case.

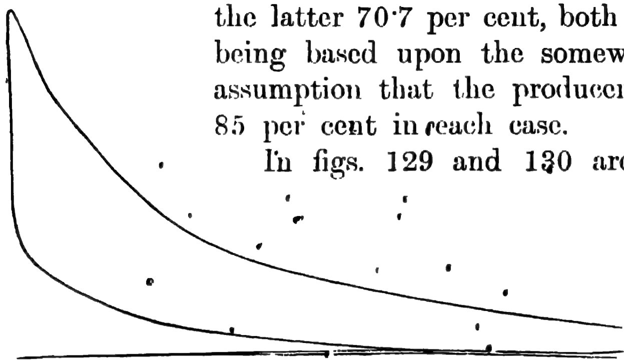


Fig. 129.—Indicator Diagram, Tangye Engine

In figs. 129 and 130 are shown two indicator cards, taken during the second test. The first diagram shows remarkably rapid and complete combustion, which is somewhat unusual with producer-gas, in which the percentage of hydrogen was found to be only 15·8 per cent, while the CO_2 percentage was as high as 6·7 per cent. The light spring diagram shows that the fluid losses during the pumping strokes were exceedingly small, and that the pressure in the cylinder had dropped to below atmospheric before the end of the exhaust stroke, showing that advantage was being taken of the inertia of the gases in the exhaust pipe to assist in withdrawing the products of combustion. This diagram also shows that the volumetric efficiency of the engine was high, for the compression line crosses the atmospheric line after

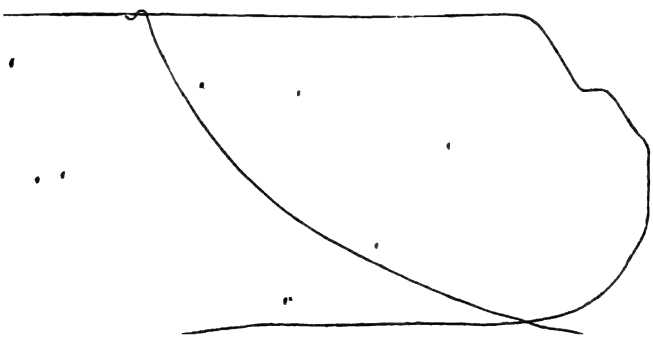


Fig. 130.—Indicator Diagram, Tangye Engine

the piston has travelled only about 6 per cent on the compression stroke. This indicates that the higher powers obtained in Test 3 must have been obtained by increasing the percentage of gas in the mixture, and not by any further increase in the valve-lift.

Ruston-Proctor Engines.—In fig. 131 is shown a sectional elevation of the Ruston-Proctor engine. This engine has a cylinder of 21·5 in. bore and 30 in. stroke, and develops 150 B.H.P. as a maximum working load when running with producer-gas at a speed of 175 R.P.M. It represents about the largest size of cylinder which can be employed without resorting to water-cooling of the piston—a condition which practically marks the limit of size for this type of engine, for water-cooling of the piston is a tiresome and difficult problem, to be avoided if possible.

It is now generally agreed that, when the power required per cylinder exceeds the maximum that can safely be obtained from an uncooled piston, it is preferable to employ a double-acting cylinder, for in this manner the power can be almost doubled for a given weight of reciprocating parts, and the cost per horse-power reduced. So long as air-cooling can be relied upon for the pistons, then the single-acting engine is the cheaper; but as soon as it becomes necessary to resort to water-cooling, it is more economical to employ the double-acting type. Single-acting engines, with uncooled pistons, are now built in sizes up to 1500 horse-power; but, in such cases, large numbers of cylinders are employed, and the output from each individual cylinder never exceeds about 160 B.H.P. In Diesel engines, the single-acting type is almost invariably employed, in spite of the necessity for water-cooling the pistons in the larger sizes, but this is because the double-acting principle introduces difficulties which do not apply in the case of the gas-engine.

In so far as the general construction and mechanical details are concerned, this engine differs very little from those which have already been described. The crankshaft is not a solid forging, but is built up, the crank-webs and balance-weights being steel castings bored out and shrunk on to the crank-pin and shaft. By this means a somewhat cheaper shaft can be employed, and the danger of balance-weights coming adrift is, of course, eliminated. This form of built-up shaft has of course been well tested in marine steam-engine practice, and has shown itself to be entirely satisfactory. The connecting-rod big-end bearing is of the marine type, with separate cast-steel housings for the bearings. The connecting-rod itself has a flat foot, and the two halves of the bearing are, of course, lined with

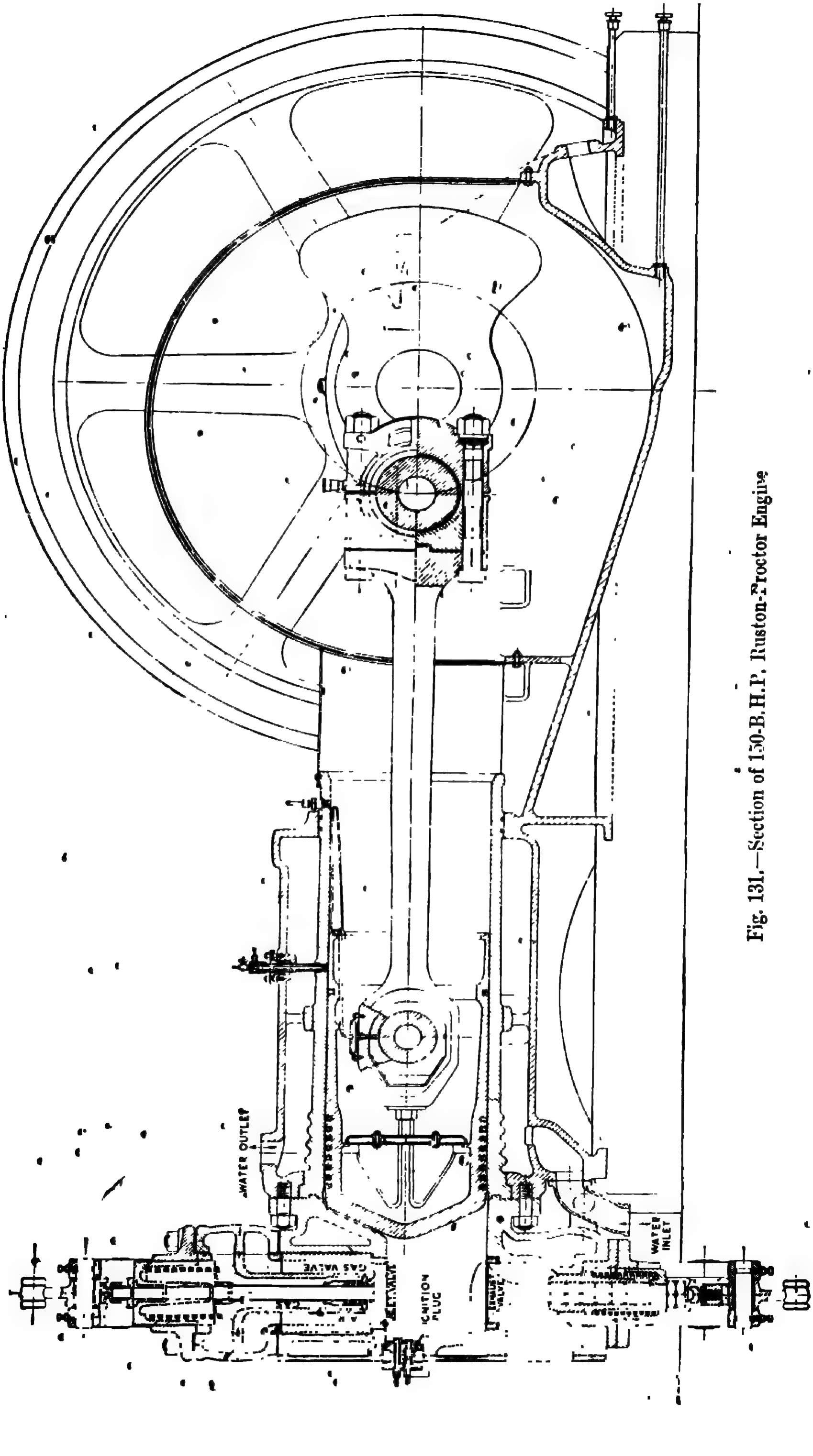


Fig. 131.—Section of 150-B.H.P. Ruston-Proctor Engine

white metal. This form of big-end bearing is less costly than the solid forged type, and is perfectly satisfactory so long as the foot of the rod is carefully spigoted into the housing of the bearing, and the big-end bolts relieved of any shearing stresses. The design of the combustion chamber is generally similar to that of the preceding engines, but it will be noticed that the water-jacket is left open at the back, and afterwards closed by separate detachable covers. This is necessary in large engines because the inner walls of the combustion chamber must be made very thick to withstand the fluid pressures, and, in consequence, their mean temperature is considerably greater than that of the cooling water. The expansion, therefore, of the inner and outer walls is by no means equal, and it becomes necessary to avoid tying them rigidly together, otherwise the irregular expansion will set up severe stresses, which may lead to cracking. This problem of the unequal expansion of the inner and outer walls of the combustion chamber or cylinder does not apply to any appreciable extent in small engines, but increases in importance as the size of the engine is increased, until, in the very large engines, it becomes the controlling factor.

The admission of gas and air is controlled in very much the same manner as in the Tangye engine. The main inlet valve carries a supplementary gas valve, and the quantity of mixture is controlled by the lift of the two valves, which, again, is under the control of the governor. The variation of lift is accomplished by providing the top of the valve with a broad flat face, over which a movable roller, controlled by the governor, travels as shown in fig. 132. This roller is, in effect, a movable fulcrum, increasing the lift of the valve as its distance from the pivot of the rocking lever is increased. The roller, of course, is relieved of all load, and is perfectly free to move during the whole period that the valve is closed. It is not, however, a truly balanced gear, because the rocking lever and valve face are not at all times parallel to one another. Consequently, there is always a tendency for the roller to be forced in one direction or the other. This tendency might react upon the governor, but is prevented from doing so by providing the roller with two conical flanges, which have the effect of binding upon the rocking-lever, and so introducing a frictional resistance. It must be noted, however, that in this case the resistance is applied only during the period that the valve is lifted, and that at other times the roller is perfectly free to move.

Both the inlet and exhaust valves are operated from a single

cam on the side-shaft, an arrangement which is decidedly neat from a mechanical point of view, and which makes it possible to place both valves in the same line, and operate them direct through straight levers. It is, however, open to the objection that it is not possible to time the opening of the two valves independently, and that there must necessarily be a considerable amount of over-

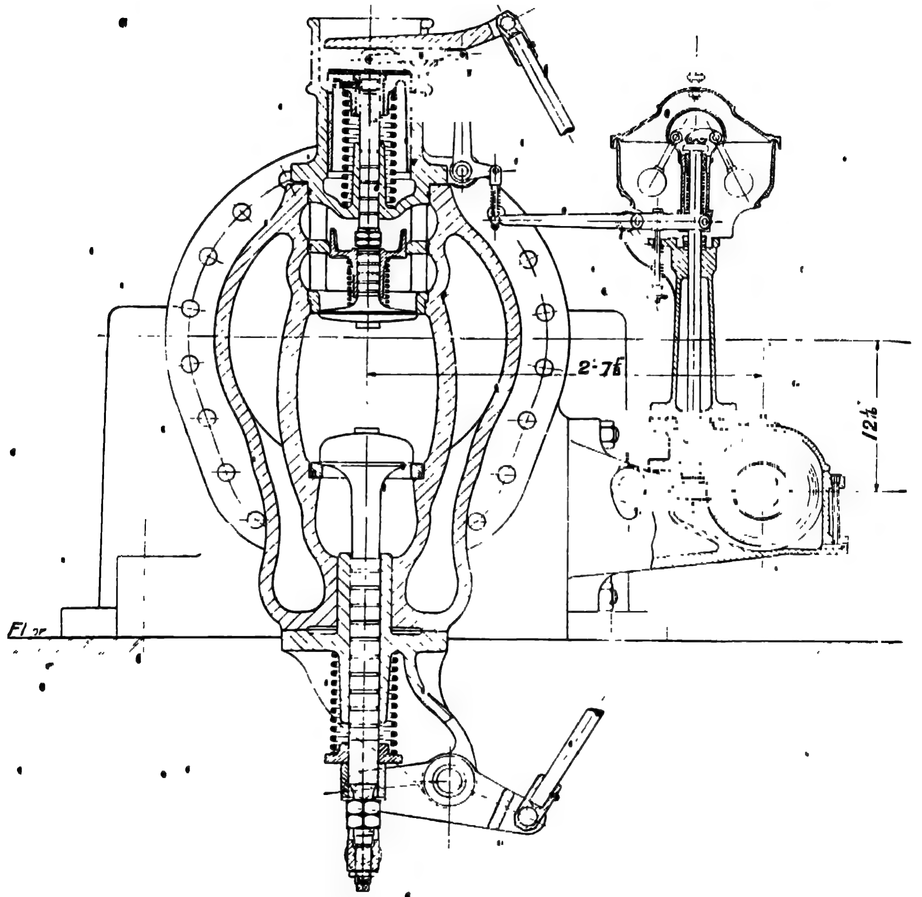


Fig 132.—Cross-section of Ruston-Proctor Gas-engine

lap, which is desirable only when the exhaust-pipe arrangements are suitable. Also, the exhaust valve cannot be opened as early as it should be, which results in an appreciable amount of fluid resistance during the early part of the exhaust stroke.

The piston is lubricated by means of a forced-fed lubricator, actuated from the side-shaft, which delivers oil to the top side of the piston. A separate sight-fed lubricator, fitted near the open end of the liner, supplies oil to a trough carried on the front end of the

piston, from which it is led by a small pipe to the gudgeon pin-bearing.

It will be noticed that the head of the piston is enclosed by means of a light detachable cover, the object of this being to prevent any oil from reaching the under side of the piston-head, where it would carbonize, and give off unpleasant smoke and smell.

The exhaust valve is very massive, both in the head and stem, with the object of maintaining an even temperature and of getting rid of as much heat as possible down the stem. It is one of the difficulties with the larger sizes of gas-engines that, in order to prevent wide variations of temperature in the working parts, with the consequent risk of both pre-ignition and failure through unequal expansion, it is necessary to make all parts which are exposed to high temperatures, and which cannot conveniently be water-cooled, very thick and heavy. In doing so, the mechanical efficiency suffers, and the wear and tear is increased, unless a very low rotative speed be employed. This, again, involves greater heat loss, and increases the cost, size, and weight of the engine.

The Ruston-Proctor engine runs normally at a speed of 175 R.P.M. per minute, corresponding to a piston speed of 875 ft. per minute, which is high for so large an engine of this type. The brake mean pressure is 72 lb. per square inch, and taking the mechanical efficiency as 85 per cent, the indicated mean pressure will be approximately 85 lb. per square inch, certainly a high figure for so large an engine with uncooled piston and valves. Such engines cannot run with very strong mixtures, owing to the high temperatures involved, and the risk of pre-ignition from overheating of the uncooled parts. To obtain a mean effective pressure of 85 lb. per square inch with a weak mixture of producer-gas and air, indicates that the volumetric and thermal efficiencies must be very high.

It is by no means an easy matter to measure accurately the quantity of gas consumed by any gas-engine, because the pulsation in the pipe-work, caused by the intermittent suction of the engine, interferes with the accurate working of any gas-meter; hence the consumption of large gas-engines using producer-gas is almost always measured on the quantity of fuel consumed in the producer. From a commercial point of view this is all that is required, but it does not give any clue as to the actual thermal efficiency of the engine alone, for the efficiencies of different producers vary considerably, both between themselves and according to the class of fuel used. In consequence of this difficulty, actual figures as to the thermal

efficiency of large gas-engines are not easily obtained, except in laboratories equipped with special apparatus for measuring the flow of gases.

Engines similar to the one illustrated in fig. 132 are built with one, two, or four cylinders, giving 150, 300, and 600 B.H.P. In all cases the cylinders are arranged side by side. In the case of two-cylinder engines, the two crank-pins are almost invariably in the same plane, so that there is one impulse at every revolution, and the interval between the impulses is the same. This arrangement provides for a uniform turning moment and good governing; but, as pointed out previously, the rotary balance is exceedingly poor. For slow-running stationary engines, however, for which ample foundations can easily be provided, this is not a very serious matter, and experience has shown that, for such engines, even turning movement and accurate governing are of more importance than rotary balance.

In fig. 133 is shown a photograph of a two-cylinder 250-B.H.P. Ruston-Proctor engine. In so far as the mechanical details are concerned, this engine resembles the one just described in all respects except that a solid forged-steel crankshaft is employed, with cast-iron balance-weights attached by means of tension bolts. The leading dimensions of this engine are:—

Bore	19·5 in
Stroke	
Number of cylinders	
Piston area	298·6 sq in.
Swept volume (cubic feet per cylinder) ...	4·67.
Compression ratio	5·4 : 1.
Maximum B.H.P.	250.
R.P.M.	185.
Piston speed	832·5 ft. per minute.
ηp (brake mean pressure)	66·2 lb. per square inch.
Diameter of inlet-valve port	7·5 in.
Lift of inlet valve	1·375 in
Effective area of opening	32·3 sq in.
Diameter of exhaust port	7·0 in.
Lift of exhaust valve	1·625 in.
Effective area of opening	35·8 sq. in.
Ratio of piston area to inlet area	9·25 : 1.
Weight of piston	850 lb.
Weight of connecting-rod	897 lb.
Weight of reciprocating parts	1220 lb.
Weight of reciprocating parts per square inch of piston area	4·1 lb.

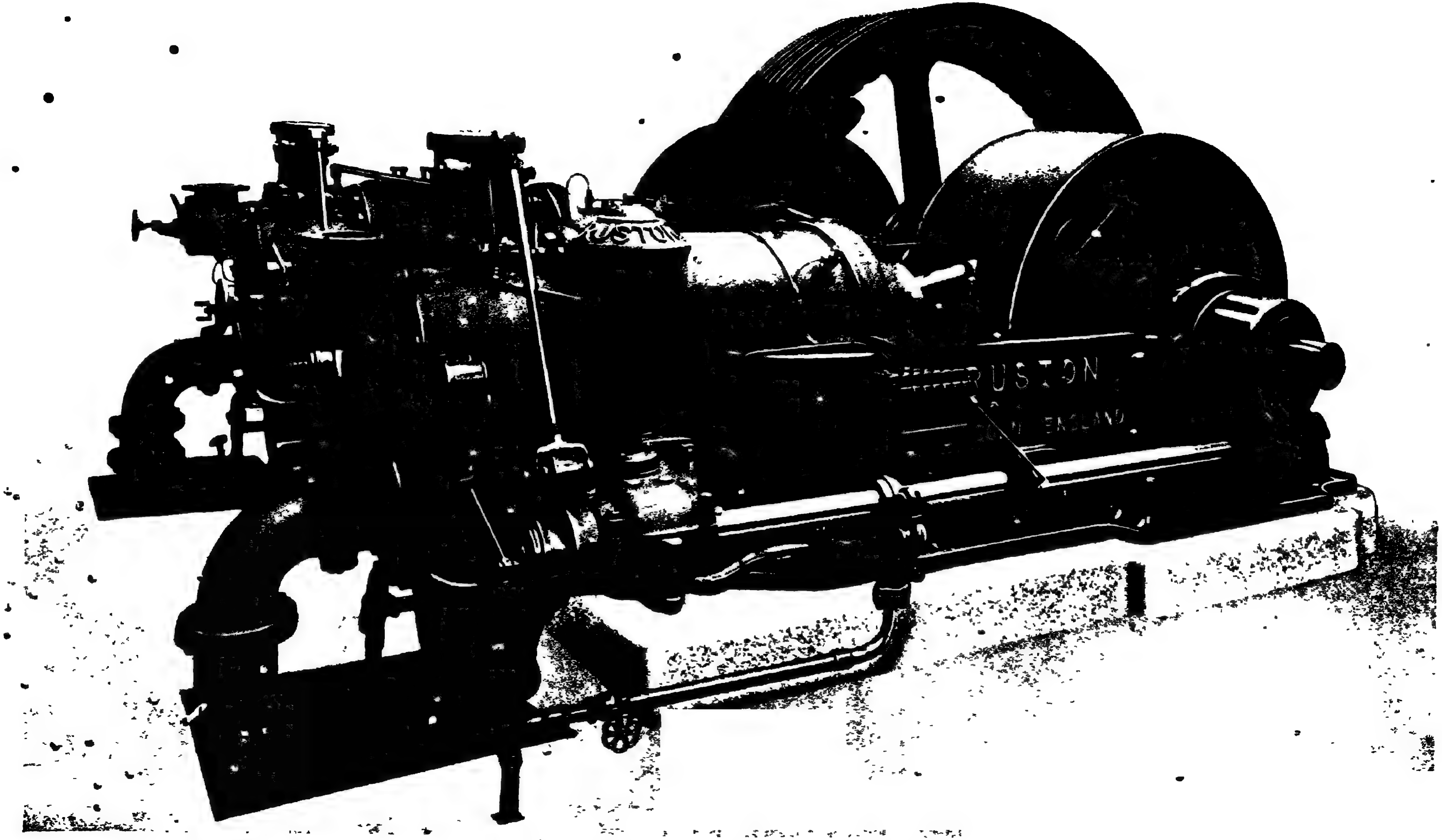


Fig. 133.—250-H.P. Ruston-Proctor Gas-engine

In actual operation, the makers state, this engine consumes 13 cu. ft. of town gas, having a lower calorific value of 650 B.T.U.s per cubic foot per B.H.P. hour; or 0·7 lb. of anthracite, having a calorific value of 14,500 B.T.U.s per pound, with a producer efficiency of 83 per cent. In either case the brake thermal efficiency is almost exactly 30·2 per cent. The air standard efficiency for this engine being 49·2 per cent, the performance is an excellent one. The makers state that the mechanical efficiency, as ascertained by indicator diagrams, is 82 per cent, which seems very low. This figure makes the indicated thermal efficiency $\frac{100}{82} \times 30{\cdot}2$ per cent $= 36{\cdot}8$ per cent, and the relative efficiency $\frac{36{\cdot}8}{49{\cdot}2} = 75$ per cent, which is altogether too high. The relative efficiency in such an engine as this could hardly be higher than 71·5 per cent, corresponding to an indicated thermal

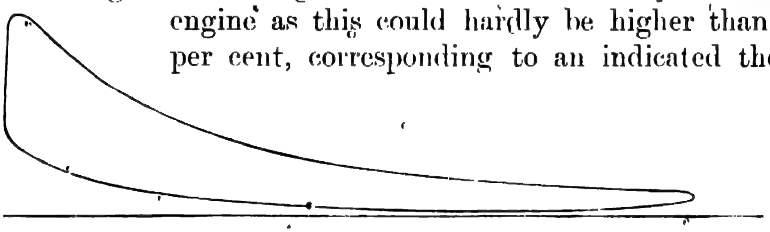


Fig. 134.- Indicator Diagram, Ruston-Proctor Gas-engine

efficiency of 35·2 per cent, and a mechanical efficiency of 85·7 per cent. The makers further state that the brake readings and the measurements of gas consumption were recorded with an accuracy which leaves little room for doubt. The indicated horse-power, however, was arrived at by means of a pencil indicator, which is by no means accurate, and generally has a tendency to read too high.

By calculation, the mechanical efficiency works out as follows:—

Fluid Loss.—The gas velocity through the inlet valve is

$$\frac{9{\cdot}25 \times 832{\cdot}5}{60} = 129 \text{ ft. per second,}$$

and that through the exhaust is slightly lower; but the opening of the exhaust valve is late, as is shown by the indicator diagram (fig. 134), so that the fluid losses may be as high as 4·5 lb. per square inch.

Piston Friction.—The weight of the reciprocating parts is approximately 4·1 lb. per square inch of piston area, so that the piston friction will amount to about 4·8 lb. per square inch.

Bearing Friction.—The bearing and other friction in a two-

cylinder engine such as this will probably not exceed 2·5 lb. per square inch.

The total losses, therefore, will amount to—

Fluid loss	4·5 lb. per square inch.
Piston friction	4·8 „ „ „
Bearing and other friction ...	2·5
Total losses	11·8

The brake mean pressure at normal full load is 66·2 lb. per square inch, so that the indicated mean pressure will be—

$$66 \cdot 2 + 11 \cdot 8 = 78 \text{ lb. per square inch},$$

and the mechanical efficiency becomes

$$\frac{66 \cdot 2}{78} = 85 \text{ per cent.}$$

If this figure be accepted, then the indicated thermal efficiency becomes $\frac{100}{85} \times 30 \cdot 2 = 35 \cdot 5$ per cent, and the relative efficiency 72·3 per cent, which still seems to be somewhat too high, and suggests that the mechanical efficiency must be even higher than the calculated one. In any case, the results obtained are remarkably good, and leave very little scope for improvement in any direction.

The Crossley Engine.—The Crossley 130-horse-power producer gas-engine, shown in figs. 135 and 136, is similar in most respects to the Ruston-Proctor engine just described. Engines of this type are, and have been, made in such large quantities that the design has been practically standardized. The construction throughout is simple and thoroughly substantial, and the whole engine has a neat and workmanlike appearance. This engine has a cylinder bore of 19½ in. and a stroke of 28 in., and develops a maximum of 130 B.H.P. when running at a speed of 180 R.P.M., corresponding to a piston speed of 840 ft. per minute. The brake mean pressure is 69 lb. per square inch, and the indicated mean pressure, on the assumption that the mechanical efficiency is 85 per cent, as stated by the makers, will amount to 81 lb. per square inch, rather a high figure for a producer gas-engine. The brake thermal efficiencies, as guaranteed by the makers, are as follows:—

Full load	25·3 per cent.
Three-quarter load	22·6 „
Half load... ...	18·4 „
Quarter load ...	12·7 „

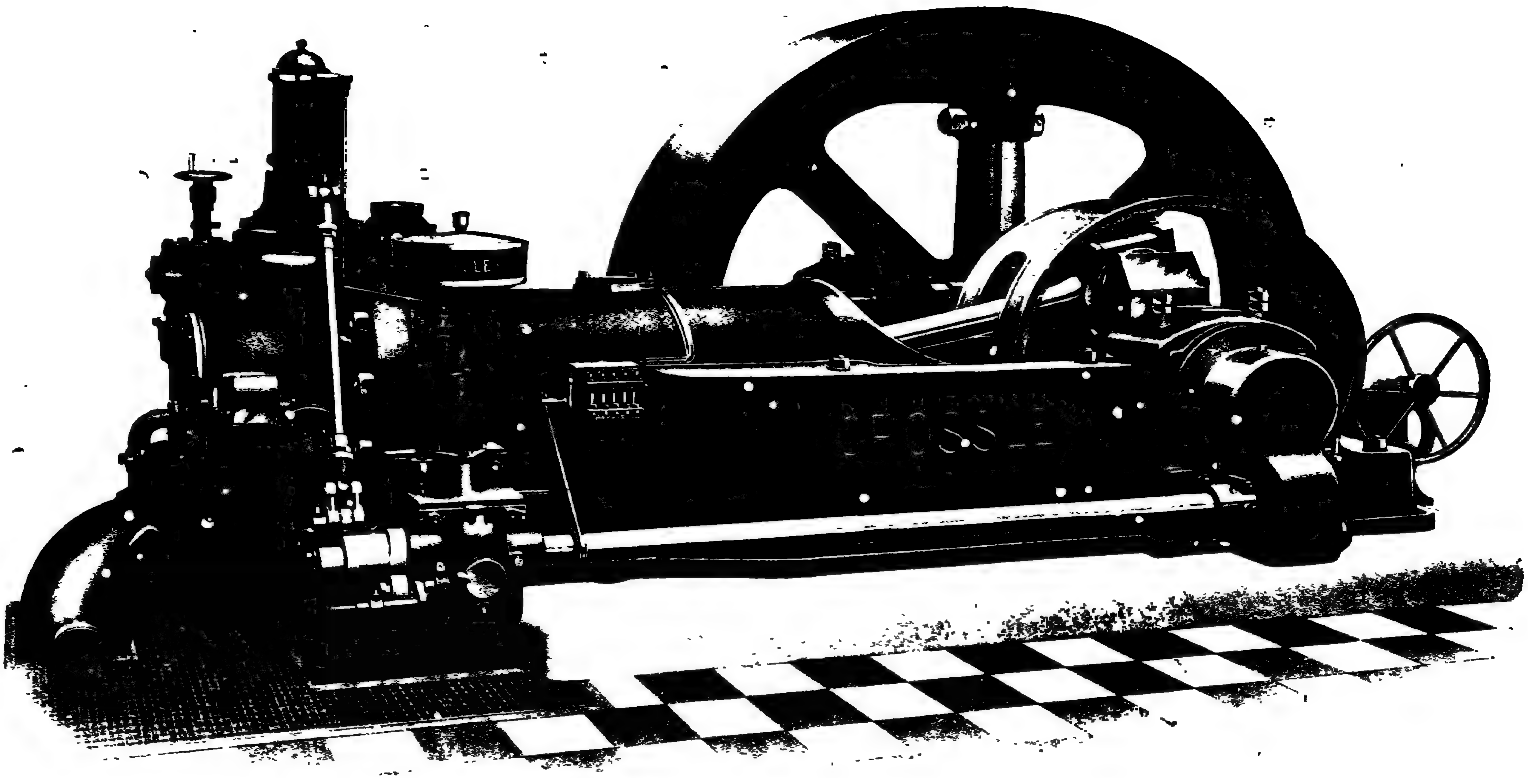


Fig 135.—1?0-H.P. Crossley Gas-engine

These figures appear somewhat poor for an engine of this class and size, but they are the figures which the makers guarantee to

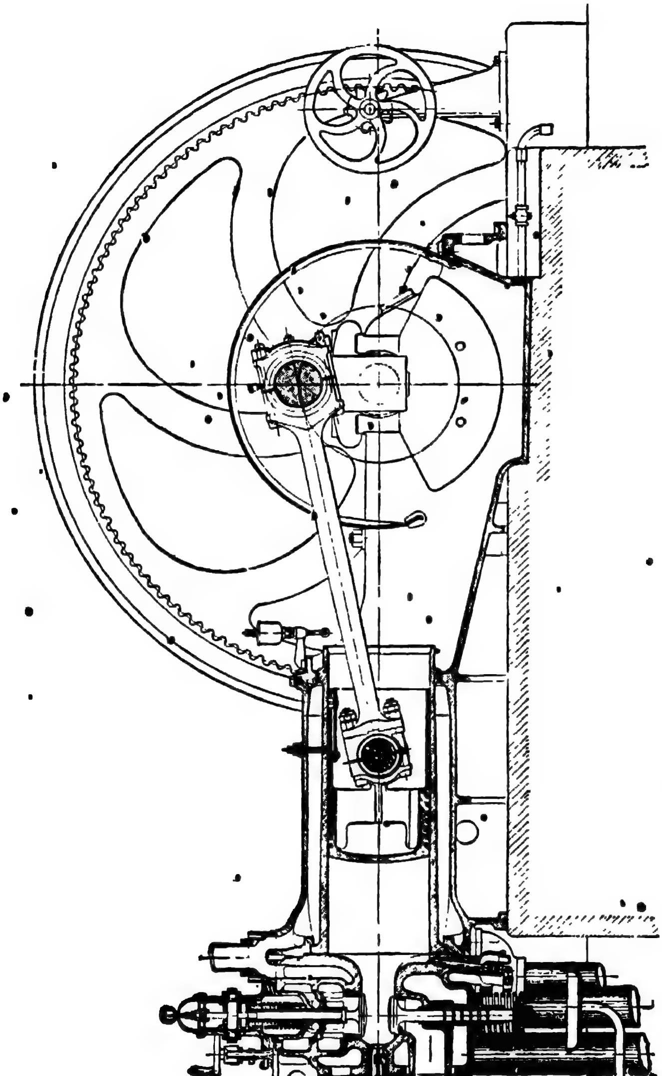


Fig. 136.—130-H.P. Crossley Gas-engine

obtain from a new engine on the test bed, and are not the best figures obtainable after careful adjustment and a considerable amount of running in. With a somewhat larger engine of 21 in. bore and 30 in. stroke, running at 170 R.P.M., and using town gas

of 550 B.T.U.s per cubic foot lower heating value, Messrs. Crossley have obtained the following results, after careful adjustment:—

B.H.P.	Gas Consumption (cu. ft. per B.H.P. hour).	Brake Mean Pressure (lb. per sq. in.).	Brake Thermal Efficiency
			Per cent.
160	15·7	76·4	29·6
120	17·0	57·2	27·3
80	19·7	38·2	25·6
40	27·7	19·1	16·8
20	41·0	9·55	11·3

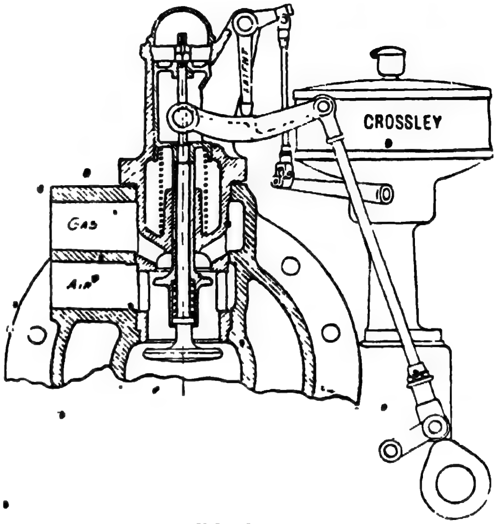
If now the mechanical efficiency of this engine, when running at a load of 160 B.H.P., be taken as 86 per cent, which is probably approximately correct for an engine of this size, running at this speed and mean pressure, the results become:

B.H.P.	I.H.P.	Mechanical Efficiency.	Mean Pressure (lb. per sq. in.).	Indicated Thermal Efficiency	Relative Efficiency.
		Per cent.	Per cent	Per cent	Per cent
160	186	86	88·7	34·1	67·5
120	146	82·2	69·5	33·2	65·2
80	106	75·5	50·5	31·3	61·5
40	66	60·5	31·6	27·7	54·5
20	46	43·5	22	26·0	51

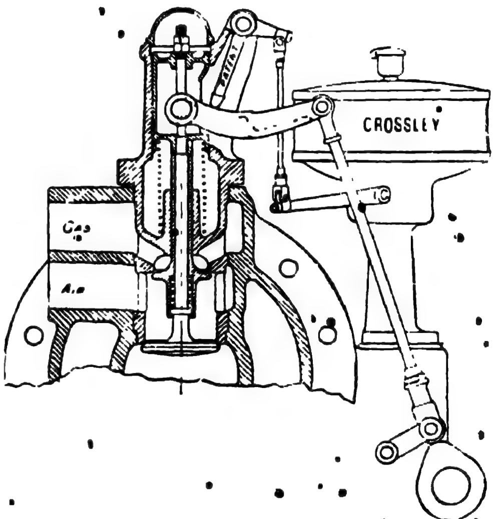
The steady fall in the indicated thermal efficiency is just exactly what one would expect in an engine such as this, which is controlled entirely by throttling both gas and air, and in which the mixture density is kept constant. There is little doubt that the indicated thermal efficiency, when running at 120 B.H.P., could be substantially improved if the mixture density were reduced, i.e. if the gas only were throttled at this load; but since, in this engine, there is no provision for stratification, the limit of qualitative governing would very soon be reached, and it is probably hardly worth while to complicate the system of governing for the sake of a small improvement in the efficiency over a comparatively narrow range of load.

The engine is governed by varying the lift of the main inlet valve, which admits both gas and air, the admission of gas being controlled by a small valve mounted concentrically on the stem of the main inlet valve, as in the Ruston-Proctor engine. The method of varying the lift is clearly shown in the cross-section of the breech-end (fig. 137). It will be seen that a curved lever is employed, one

end of which is pivoted to the inlet-valve stem, and the other to the push-rod actuated by the cam. The fulcrum of this lever is the end of a small radius rod, pivoted above the centre of the lever, actuated by the governor. The radius rod is not in contact with the lever while the valve is on its seat, and it can, therefore, be moved by the governor without friction. It is obvious that the nearer the fulcrum is brought to the inlet valve the smaller is the lift of this valve, and therefore the smaller the quantity of charge taken into the cylinder per cycle. From a mechanical point of view this arrangement is excellent, for it is perfectly balanced and frictionless, and in practice it certainly works admirably.



Full load position



Light load position

Fig. 137 —Governing Mechanism of Crossley Engine

The two indicator cards shown in fig. 138 were taken from this engine, one when running on nearly full load, and the other dead light. The light-load cards, however, vary so greatly from one cycle to the next that very little information can be obtained from them. The full-load card shows very rapid combustion, indicating considerable turbulence within the cylinder. The compression pressure is approximately 150 lb. per square inch, and the maximum pressure about 390 lb. per square inch.

The inlet valve, together with the gas valve and seating, are all mounted in a separate cage, and may easily be withdrawn for inspection or cleaning. The exhaust valve is of cast iron throughout. This involves the use of a very heavy stem; but such

a stem is necessary, in any case, in an engine of this size, in order to conduct the heat away from the centre of the valve. Cast iron withstands the high temperatures better than any other material, and is less liable to pitting or corrosion at the point where the gases impinge upon the stem. The main objection to its use is, of course, its low tensile strength, but in a slow-running engine, such as this, the inertia of the valve stem is not a very serious matter; and there can be no objection to its use, provided that the cam be designed to permit of gradual closing, and that the valve spring is sufficiently strong to prevent the valve from jumping. The exhaust valve-seating is of hard cast iron pressed into position, thus giving better wearing properties, and permitting of easy renewal. The mechanical features of this engine resemble those of the Ruston-Proctor, Tangye, &c., so closely that it is not worth while to investigate them in detail.

The piston, gudgeon-pin, exhaust-valve stem, and the centrifugal oil-ring for the crank-pin bearing are all supplied with oil under pressure from small oil-pumps driven by an eccentric from the side-shaft. These pumps deliver oil in very small quantities as required, and do not circulate the oil, as is usual in enclosed high-speed engines. The main- and side-shaft bearings are lubricated by means of oil-rings which dip into a bath of oil below the bearings, in the usual manner.

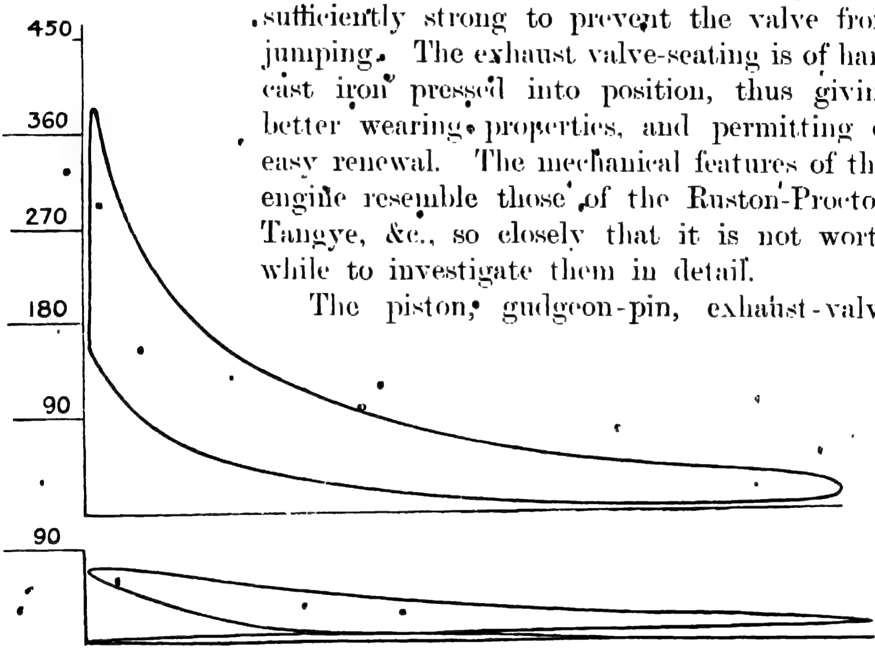


Fig. 138.—Indicator Diagrams, Crossley Engine

Starting is effected by means either of compressed air, supplied by a small air-compressor and stored in a receiver, or by pumping into the cylinder a charge of petrol or coal-gas, and igniting it by tripping the magneto by hand. In either case the fly-wheel must be barred round to the correct starting position.

CHAPTER XXIII

VERTICAL GAS-ENGINES

The engines previously dealt with may be regarded as typical of the larger sizes of single-acting horizontal gas-engines, and, although there are numerous other makers of the same type of engine, the differences in design are so slight as to be hardly worth considering. In the design of engines of the vertical type, there is considerably more variety to be found. Vertical engines are almost always intended to be run at higher rotative speeds, hence they all possess certain features in common, such, for example, as forced lubrication, enclosed crankcases, and throttle governing. They are not, as a general rule, quite so efficient as the horizontal type, probably because they generally employ a short stroke. In England it is becoming common practice in large vertical engines to use tandem single-acting cylinders, an arrangement which has many good points. For example:—

1. With two single-acting cylinders in tandem there is an impulse every revolution of the crank.
2. The reciprocating weight, per cylinder, is reduced, hence, higher mechanical efficiency.
3. The cost is reduced, for it is obviously much cheaper to add one cylinder, one above the other, than alongside, where it would require a separate connecting-rod and crank.
4. The whole engine is very compact, especially when, for the sake of large power, a very large number of cylinders must be employed.
5. By closing the bottom end of the upper cylinder, an air-buffer can be formed which will counteract, or go far towards counteracting, the inertia of the reciprocating parts, on the downward stroke, while the inertia on the upward stroke is cushioned by the compression in one or other of the two cylinders.

With the tandem arrangement, the minimum number of cranks which can be employed (to give any semblance of balance) is two,

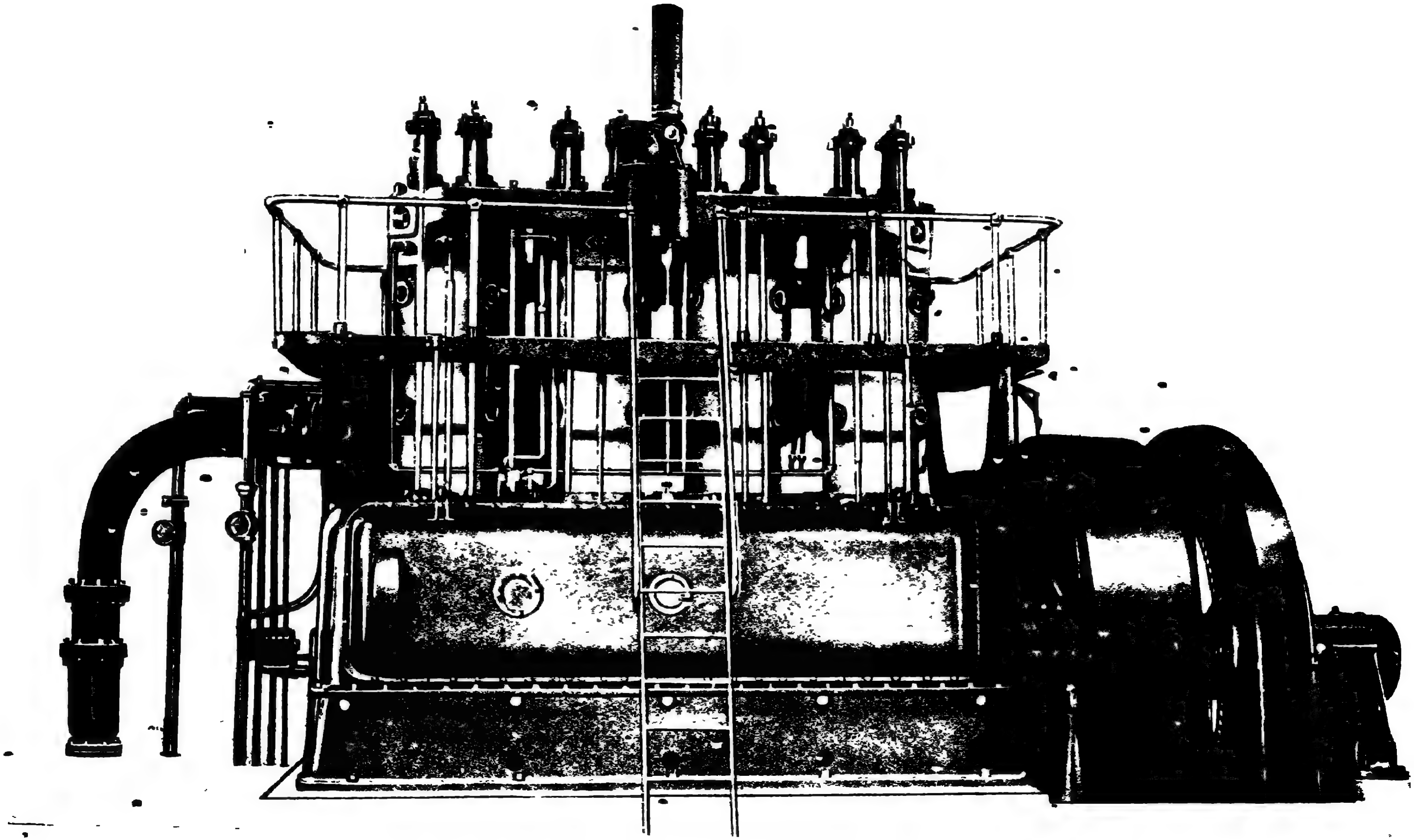


Fig. 139 —Four-cylinder 500-H.P. $21\frac{1}{2}$ in. × 24 in. 200-R.P M. Rathbun-Jones Producer-gas Engine

and four cranks are necessary to obtain a good rotary balance. Consequently, such engines cannot be built with less than four cylinders, and generally have six, eight, or even twelve, the latter number providing for as much as 1500 B.H.P. without resorting to water-cooling of the pistons.

The Rathbun Engine. — Of the ordinary type of vertical engine, that illustrated in figs. 139 and 140 may be taken as an average example of the best class. This is an American engine, made by the Rathbun-Jones Company, of Toledo, Ohio, who build a large number of vertical engines, ranging from 25 to 125 horse-power per cylinder.

The Rathbun engines are designed to run on either producer-gas or natural gas. The latter is a very rich fuel, consisting almost entirely of methane and ethylene, and has a lower heating value of about 1000 B.T.U.s per cubic foot. The engine illustrated has four cylinders, and develops 500 B.H.P. when running at a speed of 200 R.P.M. The bore is $21\frac{1}{2}$ in. and the stroke is 24 in., corresponding to a piston speed of 800 ft. per minute.

Referring to the sectional drawing, it will be observed that no liner is employed. This is usual in vertical engines, in which the cylinders are always cast separately from the base-chamber, and can therefore be renewed bodily almost as cheaply as a separate liner. The combustion-head is of slightly domed formation, and carries both the inlet and exhaust valves, which are mounted vertically. Separate detachable seatings are used for both valves, so that they can be easily withdrawn for inspection, or, if necessary, for grinding.

Fig. 141 shows sections of the combustion head, from which it will be seen that ample water-cooling is provided around and between the valve seatings, and that the thickness of metal throughout the head is very uniform.

The valves are operated from eccentrics mounted on the side-shaft in the base-chamber, through the medium of long push-rods and rolling levers, as shown in fig. 142. The exhaust valves are hollow, very light, and water-cooled, the water being led to and from the valve spindles by means of flexible tubes. By using water-cooling, a lighter valve can be fitted, the noise and wear and tear of the valve gear is reduced to a minimum, and a higher compression employed. On the other hand, water-cooling of the exhaust valves introduces a number of mechanical difficulties, and, in the event of failure of the water circulation, pre-ignition and distortion are bound to occur.

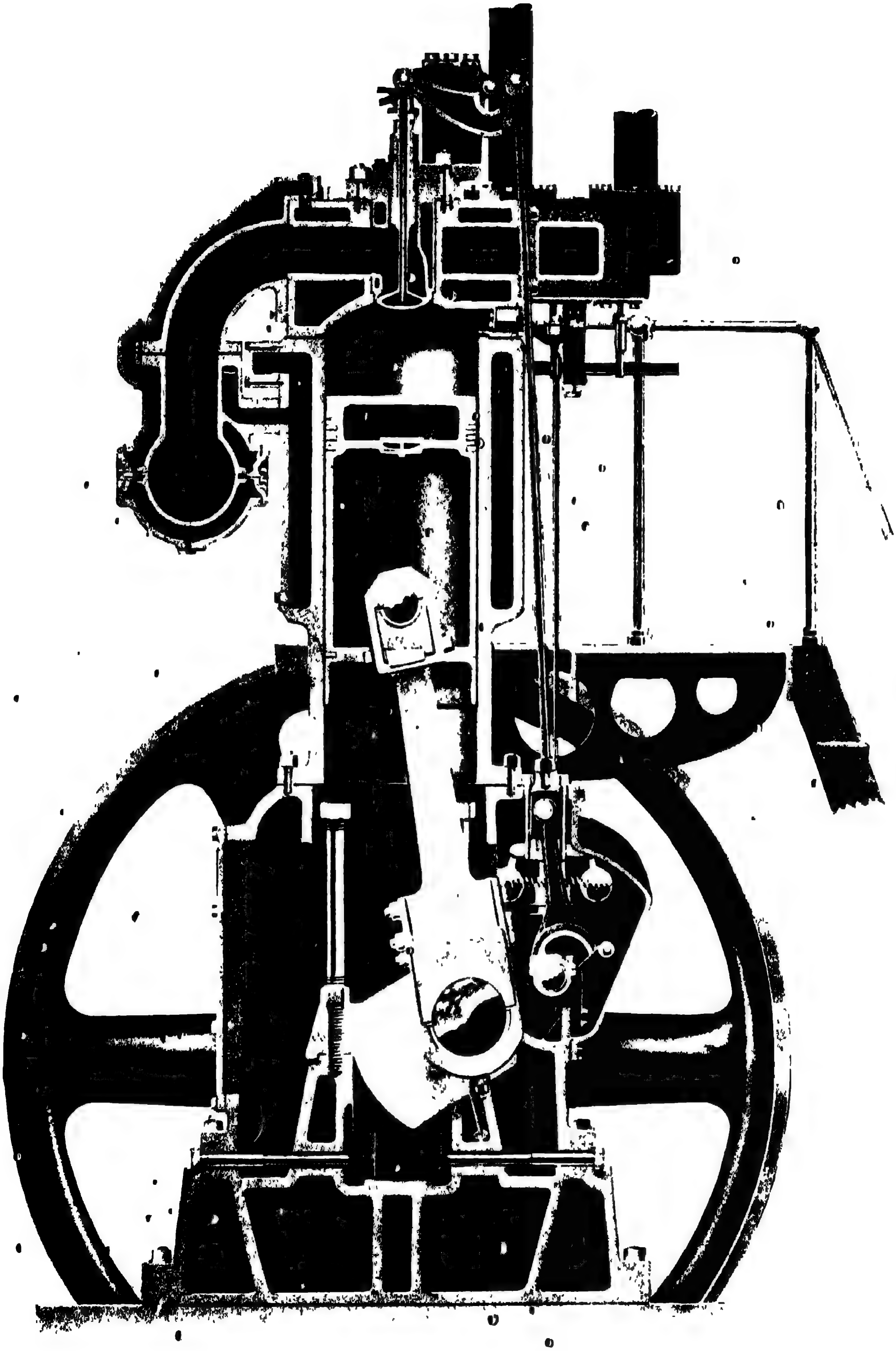
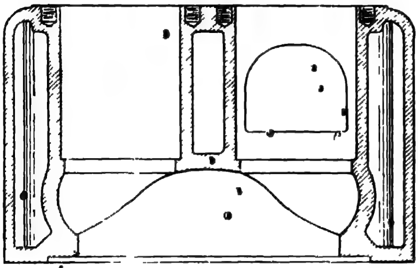


Fig 140. Rathbun Jones Gas-engine

The pistons are very long, and are provided with five rings. The upper ring is of the ordinary spring type, but the remaining four are of a special construction, designed with a view to maintain contact always against the lower side of the slot, and so prevent oil from passing up the piston into the combustion chamber, where it would carbonize. Another very excellent and important feature about these pistons is that provision is made to allow of the free escape, through a port in the cylinder walls, of any gas that may succeed in passing the piston-rings. By this means, not only is the lubrication of the piston greatly improved, but any risk of burnt gases, often containing sulphur or



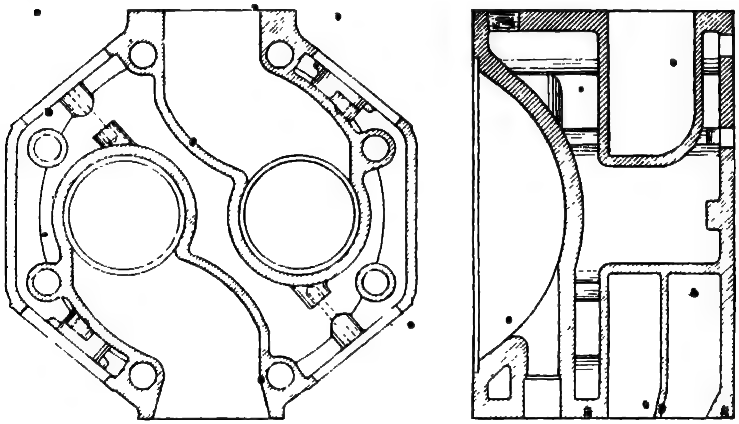


Fig. 141.—Rathbun-Jones Gas-engine, Details of Cylinder Head

other corrosive constituents, passing down to the enclosed crank-chamber, is avoided. The top of the piston is partitioned off, as in the Ruston-Proctor engine, in order to avoid carbonization of the oil on the hot under side of the piston-head. This is particularly important in an enclosed engine, for any carbon so formed is liable to fall back into the base-chamber, and seriously interfere with the lubrication.

The crankshaft is carried in white-metal-lined bearings, of which both the upper and lower halves can be adjusted, the latter by means of a wedge which can be moved sideways from outside the

crankcase, by means of a long screwed bolt. This arrangement also permits of the complete removal of any bearings without disturbing the crankshaft. The studs holding the top halves of the bearings are carried up to the top of the base-chamber and materially assist in stiffening it; at the same time they make it possible to fit very large inspection doors. The connecting-rod is of forged steel with a separate steel strap encircling the big-end bearing, and adjustment is provided for by means of a tapered collar, as in a locomotive connecting-rod. The bearings themselves are light, malleable-iron shells, lined with white metal. The small-end or gudgeon-pin bearing is adjusted by means of a wedge in the same manner as the main bearings. The bearing surfaces are of phosphor bronze, and the gudgeon-pins themselves are of mild steel, case-hardened and ground.

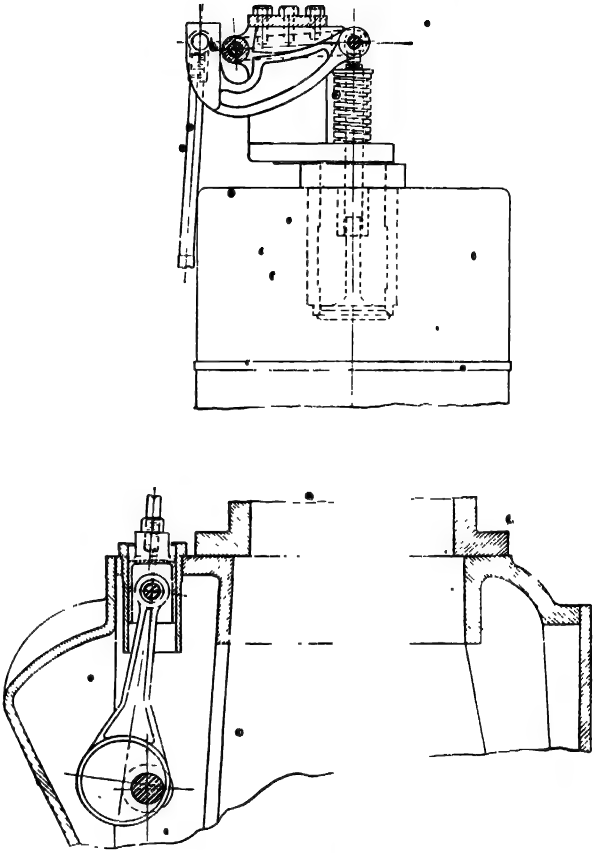


Fig. 142 --Roller-lever Mechanism for Operating Valves on Rathbun Gas-engine

A centrifugal governor is provided, enclosed within the base-chamber, and driven from the side-shaft by means of bevel gearing. It operates upon a balanced disk valve consisting of two disks, one admitting air and the other gas. These are so proportioned that the density of the mixture is slightly increased, when running on light loads, to compensate for the larger dilution with exhaust products. The governor also operates upon the timing of the ignition by slightly shifting the position of the contact-maker on the high-tension magneto, so that ignition takes place early in the stroke, as the load is reduced. This again com-

pensates for the slower burning of the mixture at light loads, due to the larger proportion of inert gases. The curve shown in fig. 143 gives the fuel consumption of this engine in terms of B.T.U.s per brake horse-power. The full line curve is the average obtained from a large number of tests, and the dotted line the best individual test.

The average brake thermal efficiency ranges from 18·2 per cent at one-third load to 26 per cent at two-thirds load, and 28·4 per

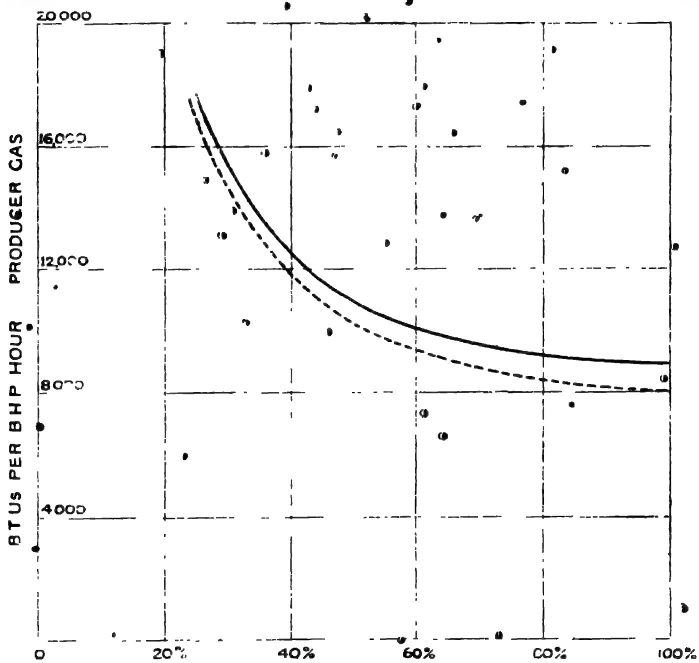


Fig 143.—Efficiency Curve of Rathbun Gas-engine

cent at full load. The best results are 19 per cent, 28·8 per cent, and 31·7 per cent respectively.' The fuel used, in each case, was producer-gas having a calorific value of about 140 B.T.U.s per cubic foot. The brake mean pressure when developing 500 B.H.P. at 200 R.P.M. is only 57 lb. per square inch. If the mechanical efficiency be taken as 85 per cent, then the indicated mean pressure becomes 67 lb. per square inch. This comparatively low mean pressure is due probably, in part, to the small size of the valves, which this form of combustion-head necessitates, but mainly to the fact that, with a light uncooled piston of 21½ in. diameter, it is unsafe to work at a higher average pressure owing to the high temperatures involved. A higher pressure would probably result in pre-ignition, and would

incur a grave risk of cracking the piston, owing to the wide range of temperature between the centre and outer edges. With such a low mean pressure as this, the mechanical efficiency could hardly be higher than 85 per cent, in which case the indicated thermal efficiency during the best test at full load will be 37·2 per cent, a remarkably high figure. The compression ratio is not stated, but it could not well be higher than 6·4 : 1 for producer-gas, even though the exhaust valves are water-cooled. For this compression ratio the air standard efficiency is approximately 52·3 per cent, and the relative efficiency 71·3 per cent, a splendid result.

Generally speaking, the Rathbun engine must be regarded as one of the best of its type.

Campbell Vertical Engines.—In fig. 144 is shown a photograph, and in fig. 145 a sectional elevation, of a four-cylinder vertical engine by the Campbell Gas-engine Company, of Halifax, which is typical of the conventional design of vertical gas-engine. The four cylinders are each cast separately, and mounted on an enclosed cast-iron base-chamber. The cylinder-heads are separate, and are simply flat water-cooled plates which can readily be removed without disturbing any other part. The inlet and exhaust valves are fitted in a side-pocket which is cast integral with the cylinder body. The inlet valve is placed over the exhaust valve, and operated by an overhead rocker and long push-rod. This form of valve gear is undoubtedly an excellent one for vertical engines. It provides a pocket for the inlet valve, which is desirable for light-load running, and, at the same time, the area of surface exposed to the gases at the time of combustion is not unduly large, since both valves are fitted in the same pocket. The removal of the valves for grinding or cleaning is a very simple matter, for the inlet valve has a separate detachable cage and seating through which the exhaust valve can be withdrawn.

As is usual in vertical gas-engines, no separate liners are employed; and this is justifiable on the grounds that each cylinder, together with its water-jacket, is a separate casting, which is comparatively inexpensive and easily renewable. The engine speed is controlled by a governor, which acts on a balanced piston valve controlling the admission of both air and gas. The arrangements of the inlet and exhaust piping are clearly shown in the photograph.

The crankshaft, camshaft, and connecting-rod bearings are all lubricated under pressure from small oil-pumps, operated from eccentrics mounted on the crankshaft. The crankshaft is fitted with

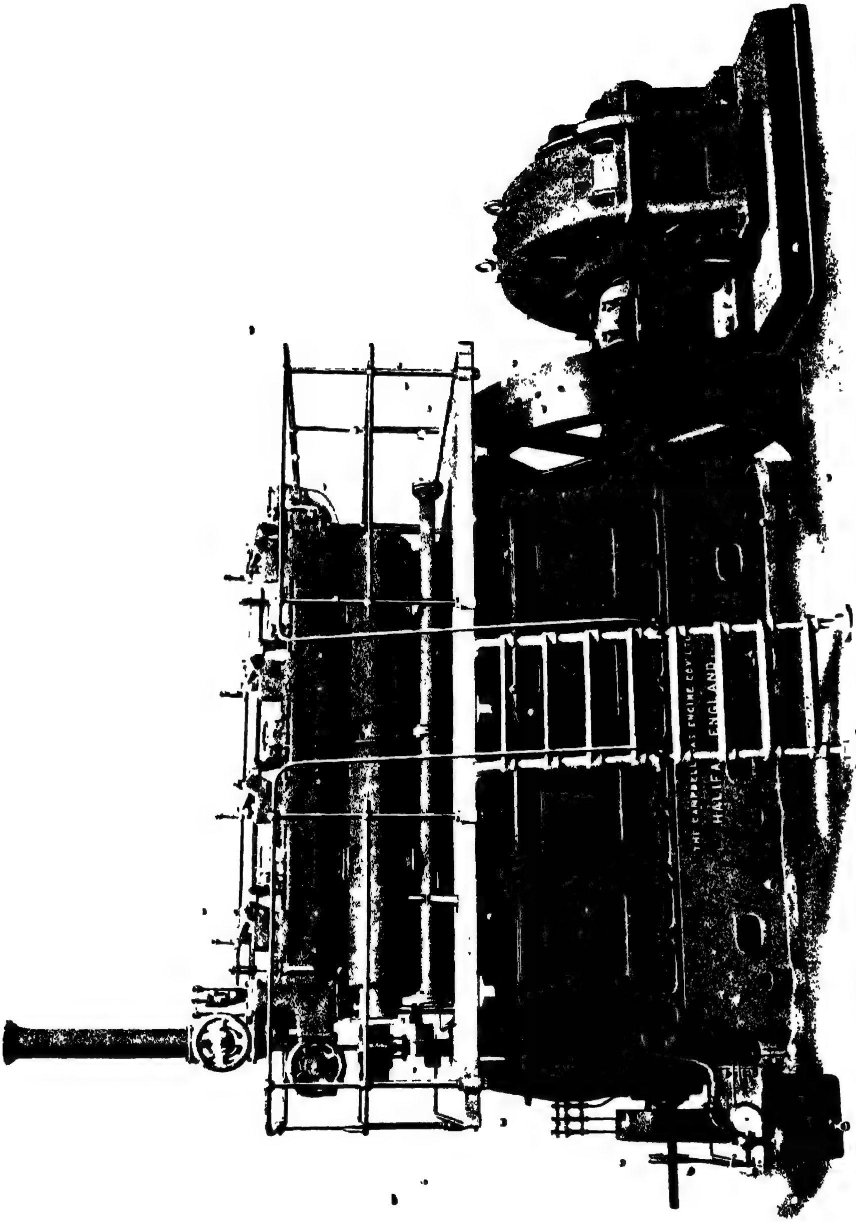


Fig. 144.—350 B.H.P. Campbell Gas-engine

heavy counterweights. These counterweights, it must be understood, play no serious part in balancing the engine, for in a four-cylinder engine such as this all primary forces are balanced. The counterweights, however, serve a useful purpose in relieving the main bearings from stresses due to the centrifugal force of the crank-pins, crank-webs, and connecting-rod big-end bearings.

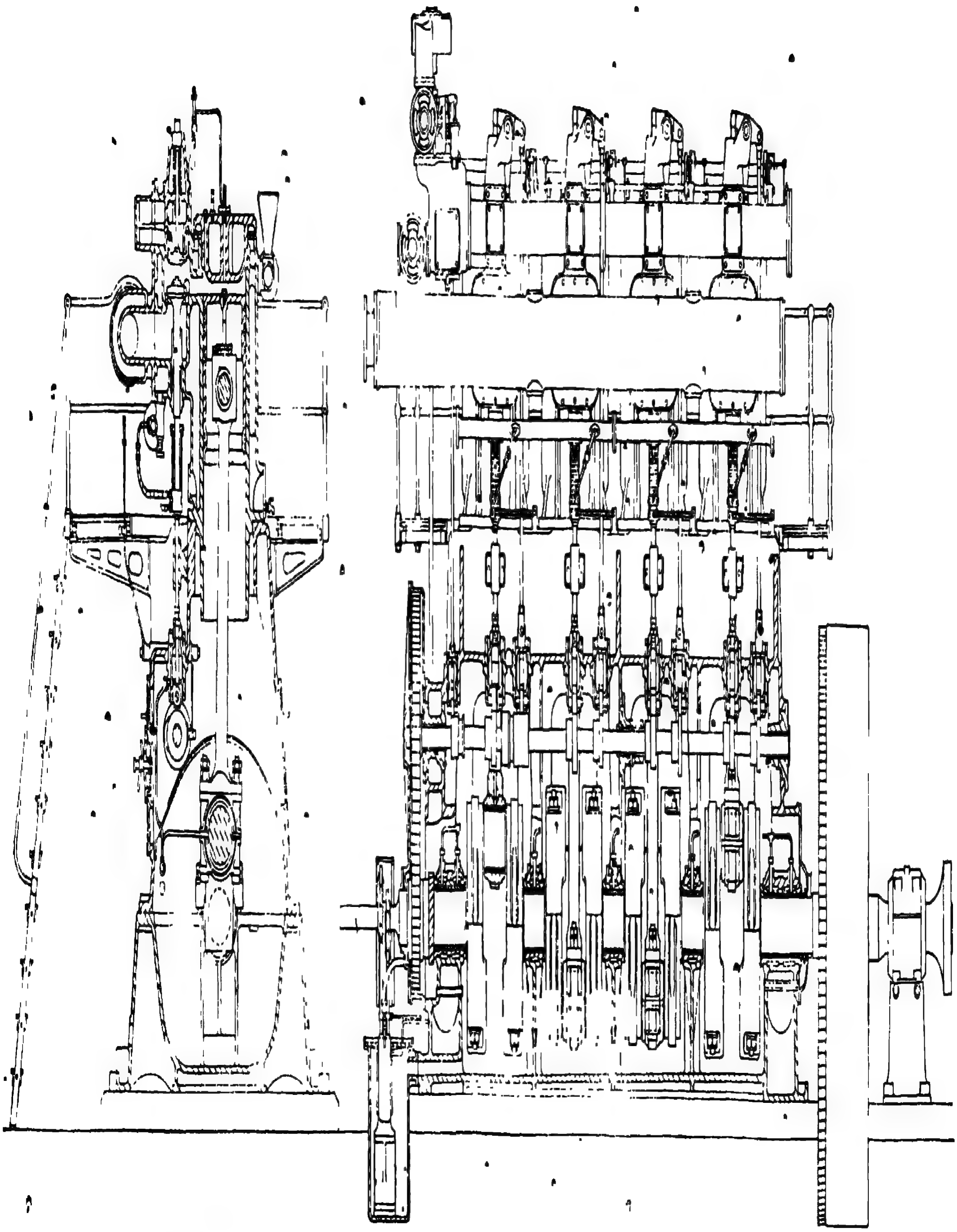


Fig. 145.—Sectional Arrangement of Four-cylinder Vertical Campbell Gas-engine. 350 B.H.P., 17 in. diameter, and 26 in. stroke

The leading dimensions of the largest type of four-cylinder vertical engine are as follows:—

Bore	22 in.
Stroke	26 in.
Number of cylinders	4.
Piston area	380 sq. in.
Swept volume per cylinder	5·72 cu. ft.
Compression ratio	5·1 : 1.
Maximum B.H.P.	575.
R.P.M.	180.
Brake mean pressure at maximum B.H.P. ..	64·5 lb. per square inch.
Piston speed	780 ft. per minute.
Diameter of inlet-valve ports	8 in.
Diameter of exhaust-valve ports	9 in.
Lift of inlet valve	1·5 in.
Lift of exhaust valve	1·75 in.
Effective area of inlet-port opening	37·6 sq. in.
Ratio of piston area to inlet area ...	10·1 : 1.
Weight of piston	785 lb.
Weight of reciprocating parts	1050 lb.
Weight of reciprocating parts per square inch of piston area	2·76 lb.
Diameter of crank-pin	12 in.
Width of crank-pin bearing	12 in.
Projected area of crank-pin bearing	144 sq. in.

Although capable of a maximum load of 575 B.H.P., the normal working load of the engine is only 480 B.H.P., corresponding to a brake mean pressure of 53·7 lb. per square inch, the limit probably being set by the pistons, which are not water-cooled. The following figures are supplied by the makers as the average test results obtained from this engine:—

Load	Brake Thermal Efficiency.
480	24·3 per cent.
360	22 ,,
240	19·5 ,,

The mechanical efficiency is estimated by the makers as 80 per cent, which will bring the indicated thermal efficiency up to—

Load.	I.H.P.	Mechanical Efficiency.	Indicated Thermal Efficiency.
480	600	80 per cent	30·2 per cent.
360	480	75 ,,	29·3 ,,
240	360	66·6 ,,	29·3 ,,

Westinghouse and National Engines.—Both the British

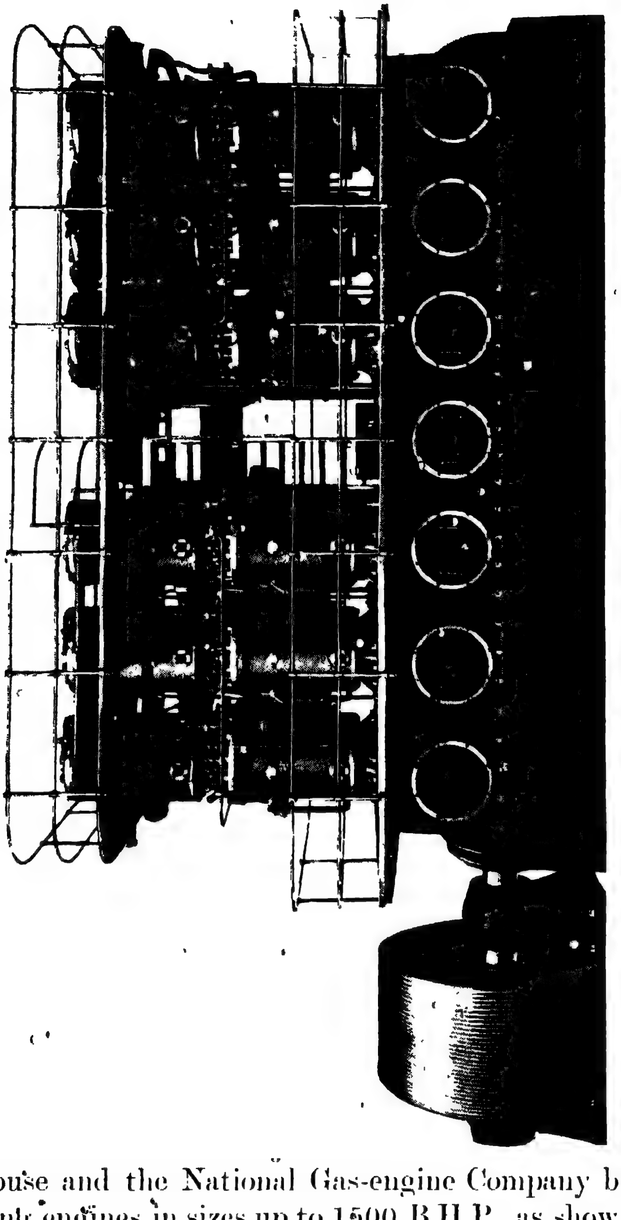


Fig. 146.—1500-B.H.P. Vertical Gas-engine

Westinghouse and the National Gas-engine Company build tandem single-acting engines in sizes up to 1500 B.H.P., as shown in fig. 146. This has twelve cylinders, each of about 22 in. bore by 24 in. stroke, and develops a maximum power of 1650 B.H.P. when running at its normal speed of 200 R.P.M. These engines, of which a considerable number have now been built, represent the largest units constructed without resorting to water-cooling of the pistons. By employing the tandem arrangement of the cylinders, each line gives one working stroke per revolution, and the inertia of the reciprocating

parts is cushioned by the compression in one or other of the two tandem cylinders. The lower piston in each case is of the ordinary trunk type, and takes the thrust from the connecting-rod in the usual manner, but the upper pistons are considerably shorter and, of course, receive no thrust. The two sets of pistons are connected together by means of a short piston-rod consisting of a steel bolt to take the tension, surrounded by a cast-iron sleeve to take the compression stresses.

Between the upper and lower cylinders a large water-cooled plug is inserted, through which the piston rod passes. No glands or packing are employed, but the piston-rod itself is provided with a number of piston-rings, which prevent leakage. With this construction, the water-cooled plug must necessarily be of greater length than the stroke of the engine. The plug projects well up into the under side of the upper piston, leaving only a comparatively small clearance space in which air is compressed during the downward stroke, and the inertia forces cushioned thereby. By this means the inertia forces are cushioned by air compression both on the upward and downward strokes, a condition that should make both for high mechanical efficiency and sweetness of running. In order to allow of fitting the plug or distance-piece between the cylinders, the bore of the upper cylinder is made slightly larger, leaving a conical seating between the two, upon which the plug rests, and against which it is held down from outside by a number of diagonal set screws, clearly shown in the illustration. By slacking back these set screws, and removing the top cover, both pistons, together with the piston-rod and distance-piece, can be withdrawn through the top of the cylinder.

Particular care has been taken in the design of the pipe-work, especially the inlet piping, in order to eliminate as far as possible the evil effects of pulsations. The inlet and exhaust valves in each cylinder are arranged and operated as in the Campbell engine, but the exhaust valves are of cast iron throughout and are uncooled. In large engines such as this, with enclosed crank-chambers, it is necessary to take very careful precautions with the ventilation of the crankcase, for there is a serious risk of the lubricating oil becoming vaporized and ignited by the high temperature of the lower pistons, and several serious accidents have occurred in cases where this has been neglected. These engines are intended to use poor gas, such as blast-furnace or producer gas, but they are also occasionally run on coke-oven gas. In this case, means are provided

for admitting a certain proportion of cooled inert exhaust gases along with the air, in order to dilute the mixture and counteract the tendency to pre-ignition, due to the high percentage of hydrogen contained in this gas. Unfortunately, no particulars are available as to the actual performance of these engines; but the author is informed, on excellent authority, that the mechanical efficiency exceeds 90 per cent, which is a truly excellent result, though by no means surprising in view of the general design of the engine. Engines of this type have become, deservedly, very popular during the last few years, and have proved themselves formidable competitors to the large, slow-running, double-acting engines, which represent the accepted type on the Continent for all large powers.

CHAPTER XXIV

LARGE GAS-ENGINES

In cases where very high powers are required, the use of uncooled pistons becomes impracticable, for it is impossible, under normal conditions, to obtain more than about 150 B.H.P. from a single uncooled piston, owing to the great difference of temperature between the centre and the circumference of the head. If any attempt is made to obtain higher powers from an uncooled piston, in any engine of normal design, the central portion of the piston is liable to reach so high a temperature as to set up premature ignition. Also, the considerable expansion of the central part introduces stresses in the material, of such magnitude that the piston is liable to fail from temperature alone, even without the application of any pressure. If it be accepted that 150 B.H.P. is, at the present time, the absolute limit of power obtainable from one cylinder, then it is clear that any increase of power can only be obtained by increasing the number of cylinders; and several firms, especially in this country, are building engines with as many as twelve cylinders, simply in order to retain the use of uncooled pistons. This multiplication of cylinders cannot be continued indefinitely, for, beyond a certain point, it introduces great difficulties with regard to the rigidity of the crankshaft and crank-chamber, and also with the equal distribution of gas to all the cylinders.

Water-cooling of the pistons is by no means an easy matter, for the water must be circulated at high pressure, and must be admitted and withdrawn through telescopic or rocking joints, all of which are liable to leak and give trouble. If the power required is so great that water-cooling of the pistons must be resorted to, then it is generally agreed that it is advisable to employ the double-acting principle, in order that the same weight of piston may receive double the number of impulses, and so reduce the ratio between the inertia and fluid pressure. By the employment of double-acting cylinders, better use is made of the material, and the cost and size of such

parts as the bedplate, crankshaft, and connecting-rod are relatively very much reduced, since they actually remain practically the same whether single- or double-acting cylinders are used. There is, of course, a considerable increase in the amount of machine work and fitting, but this, in a large engine, is more than compensated for by the reduction in the weight of material.

The whole question of single- versus double-acting engines really resolves itself into one of cost of production. So long as a plain uncooled piston can be used, the single-acting principle is the cheaper; but so soon as it becomes necessary to water-cool the pistons, then it generally becomes desirable to use the double-acting principle. In very large engines, of from 1500 B.H.P. and upwards, there is little doubt that the double-acting principle is the cheaper; but for powers of from 300 to 1500 B.H.P. it is still open to question whether it is not simpler and cheaper to employ a number of single-acting cylinders with uncooled pistons. Much, of course, depends upon the purpose for which the engine is required, and the tools, &c., at the manufacturer's disposal. For such purposes as driving electric generators, it is probable that the advantage lies with the multi-cylinder single-acting engine, because such an engine can run at a higher rotative speed, and so reduce the cost of the generator. For this purpose, single-acting engines, in powers up to 1500 B.H.P., with cylinders up to about 22-in. bore, are frequently employed. For driving blowing tubs for blast-furnaces, the double-acting engine has a great advantage, on account of its lower speed, and the use of a piston-rod which can easily be extended to carry the blower piston.

The use of large gas-engines, of 1500 horse-power and upwards, is practically restricted to iron- and steel-works, and collieries, where waste gases are obtainable. In a few cases, large producer plants have been laid down; but apparently they do not compare favourably with steam plants, when the capital cost and maintenance over a long period are taken into consideration. Unlike the steam-engine, the gas-engine does not become appreciably more economical as the size is increased. Consequently, although in small powers it is far more economical than the steam-engine, in very large powers it does not retain this advantage to anything like the same extent.

Continental Designs.—The large Continental four-cycle double-acting engines all follow the same general design, which has now become practically standardized. Such variations as there are are restricted to small details, such as the operation of the valves,

although the actual construction of the cylinder body is a point on which there is still a considerable difference of opinion. These large engines are almost invariably built in the tandem form, for the obvious reason that this form gives one impulse at every stroke. If only a single cylinder were employed, there would be two impulses, followed by two idle strokes, a condition which would necessitate the use of an enormous fly-wheel to obtain any reasonable degree of regularity. The addition of a second cylinder, in tandem with the first, does not involve any appreciable increase in the scantlings of the bedplate, crankshaft, or connecting-rod; hence, the power can be doubled with an increase of probably only about 60 per cent in the cost, to say nothing of the more regular turning moment, and the higher mechanical efficiency which the addition of a second cylinder affords.

In the very large engines it is usual to employ two tandem sets, coupled together with the fly-wheel and electric generator (if any) between the two, following the usual design of a cross-compound steam-engine. Since the design of all these large engines is so much alike, it is proposed to take one example and examine it in some detail, rather than devote much space to descriptions of the products of a number of different manufacturers. The design appears to have been originated by the Deutz Company, of Cologne, who, after building a considerable number of large double-acting engines, resorted to the smaller single-acting type, for the manufacture of which their plant is very much better equipped.

One of the first companies to realize the commercial possibilities of the Deutz double-acting engine was the firm of Erhardt & Sehmer, of Saarbrucken, Germany, who started on the manufacture of this type in 1903, and up to April, 1912, had turned out no less than 116 engines of an aggregate horse-power of 168,000 B.H.P. Messrs. Erhardt & Sehmer started on the lines of the Deutz Company; but, with the help of their very capable engineer, Herr Drave, they have succeeded in both improving upon and simplifying the original design to such a degree that the latest type of Erhardt & Sehmer gas-engine is probably the simplest and best of any that have been built.

A 2200-B.H.P. Engine.—In fig. 147 is shown a two-cylinder tandem Erhardt & Sehmer gas-engine, of 2200 B.H.P., running on blast-furnace gas at a French iron and steel-works.

In very large gas-engines, such as are now under discussion, the main source of trouble is that due to unequal expansion of the

various parts when subjected to the very high temperatures of the working fluid. This trouble is, naturally, greatest in the cylinders, which, in the first place, must be secured in such a manner that

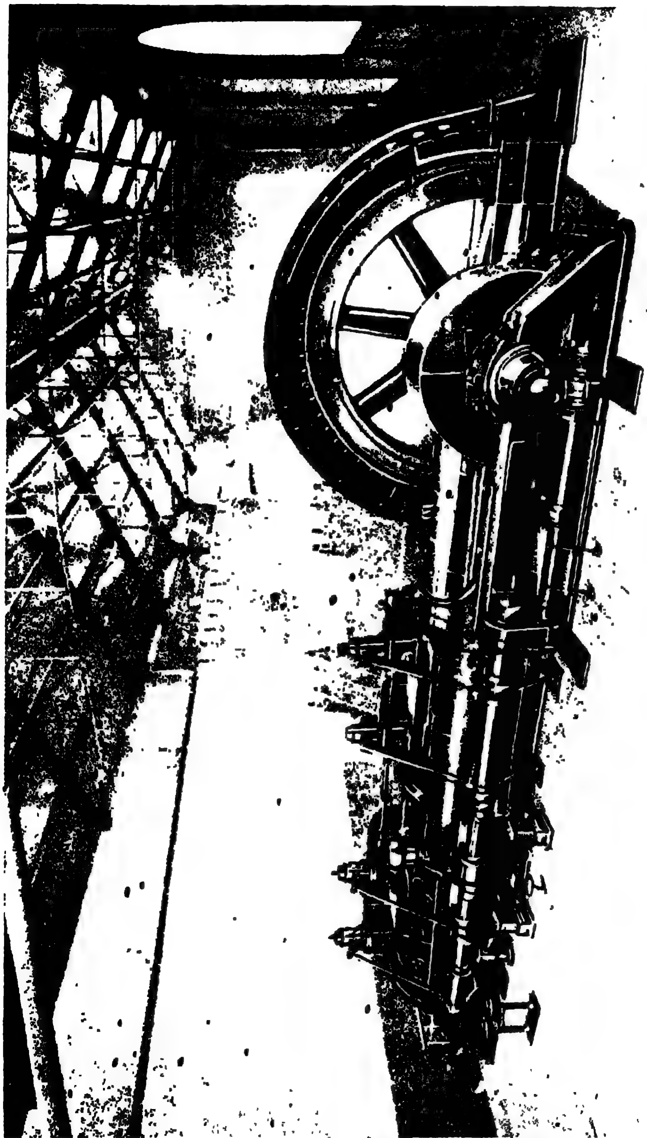


Fig. 147.—Ehrhardt & Sehmer Gas-engine, 2200 B.H.P., 1300 mm. stroke, 94 R.P.M., running on Blast-furnace Gas

they are free to expand in any direction. The great thickness of metal necessary to withstand the high pressures which occur under normal running conditions, and the even higher pressures which may be set up in the event of severe pre-ignition, results in a very considerable difference of expansion between the inner and outer

surfaces of the cylinder walls, with the result that, although the outer surfaces are in tension, the inner surface may, owing to its greater expansion, be in compression. It is obvious that, after a certain limiting point is reached, any further increase in the thickness of the walls will add nothing to their structural strength, because its value will be counteracted by the severe internal stresses set up by the unequal expansion. It is this question of the unequal expansion of the cylinder walls which really limits the size of cylinders which may be safely employed, for it limits the thickness of the walls and, therefore, the diameter of the cylinders.

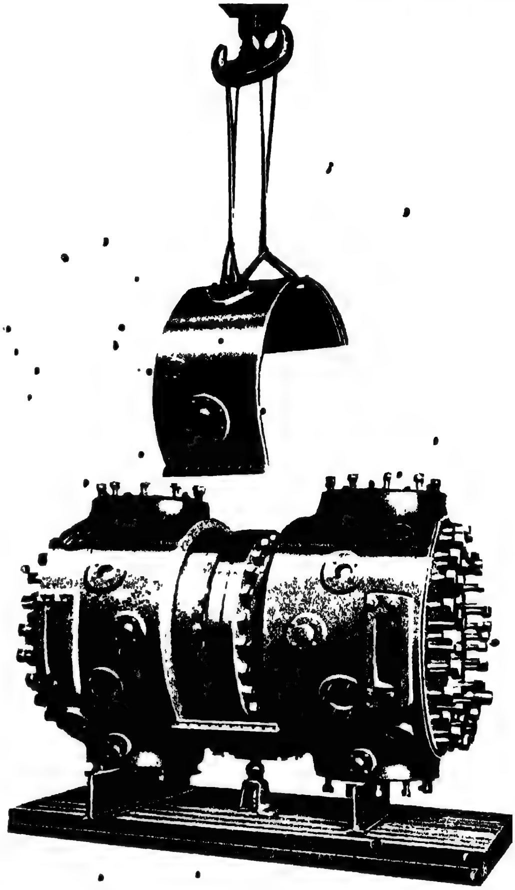


Fig. 148.—Cylinder with Jacket Casing removed

In large engines, the difference between the expansion of the cylinder barrel and the water-jacket is of quite a large order, and care must be taken to ensure that no extra stresses are thrown upon the cylinder proper by the jacket. It is in this direction that the views of various manufacturers differ considerably. In the Erhardt & Sehmer engine the problem has been dealt with in a most capable manner. The cylinder proper, which also contains the valve pockets, is constructed in two halves, bolted together round the centre, as shown in figs. 148 and 149.

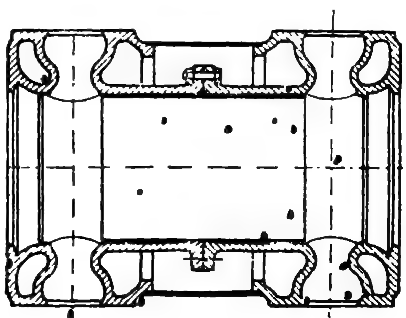


Fig. 149.—Section of Cylinder

Each half includes a portion of the water-jacket; but the central

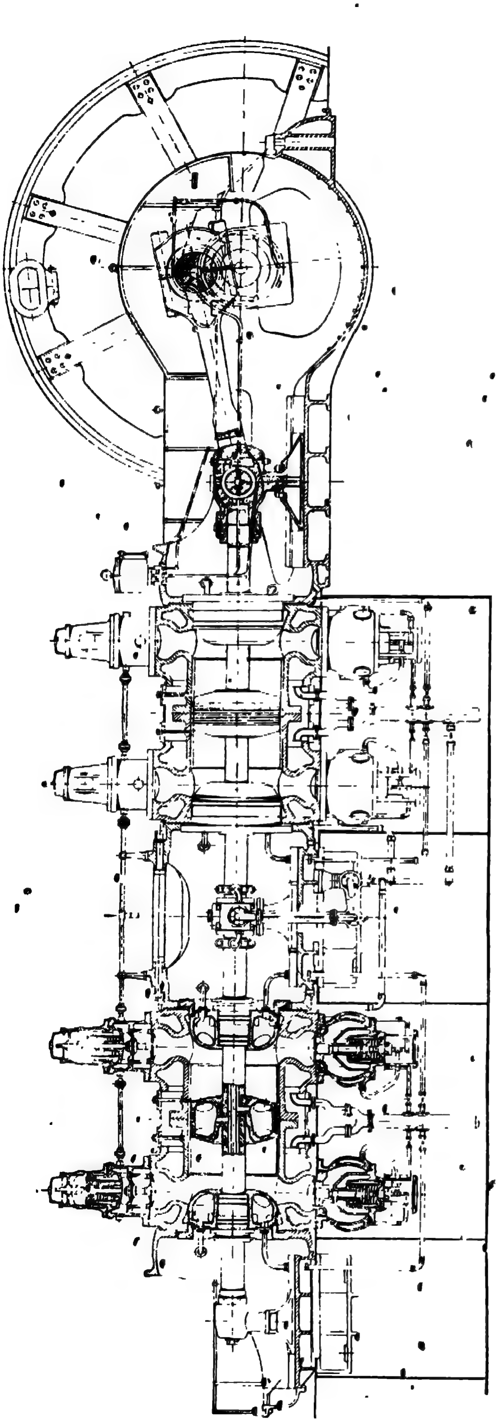


Fig. 150.—Section of an Ehrhardt & Sehmer Gas-engine. Output, 2000 B.H.P., stroke, 1300 mm., 90 R.P.M.

part is left open, so that it is free to expand in any direction, and afterwards closed by a light steel jacket, also made in two halves, and kept watertight by means of rubber rings. The cylinder liner is of hard cast iron, and is provided with a small central projecting flange, clamped between the two halves of the cylinder, so that it is held at the centre and is free to expand in either direction. The whole construction is both simple and ingenious, and permits of the free expansion of the liner and cylinder body. Great care is taken to ensure that the thickness of metal throughout the whole of the cylinder shall be uniform, and that there shall be no sharp corners or abrupt changes of section, from which cracks might easily start. The cylinder end-covers are merely plain water-jacketed plugs, of the simplest possible form, and call for no special comment.

General Design.—The general design of this engine is well illustrated in the sectional

elevation (fig. 150), which represents a two-cylinder tandem engine developing 2600 B.H.P. at a normal speed of 90 R.P.M. The engine is built up of five different sections, bolted together in a row, so that they are all free to expand longitudinally. The main frame, which is shown in fig. 151, forms the first section. To this is bolted No. 1 cylinder, followed by a distance-piece, then No. 2 cylinder, and (finally) the rear cross-head guide. The piston-rod is drilled for the passage of the cooling-water to and from the pistons, and is carried on three slippers, one of which forms the main cross-head. The pistons themselves are comparatively short, as in steam-engine practice, and do not bear upon the walls of the liner, but

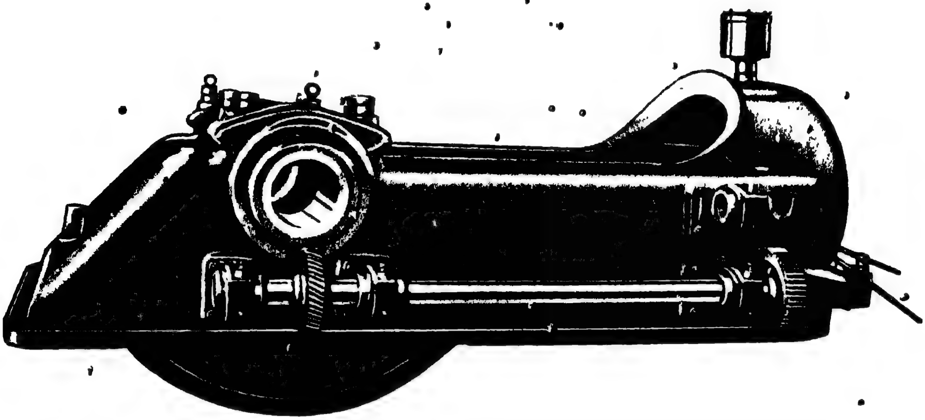


Fig. 151. —Frame with Camshaft

are supported entirely by the piston-rod, which is of sufficiently large diameter to ensure against sagging.

The valves are arranged in pockets, the inlet valves above and the exhaust valves below, and all are operated from a single side-shaft, as in the usual horizontal-engine practice. The cylinder covers are plain water-cooled disks or plugs, which can be removed without disturbing any other part of the engine.

The main frame is somewhat similar, in general design, to the frames used in ordinary single-acting horizontal engines, except that it does not include the cylinder-jacket. The main bearings are lined with white metal, and are lubricated under pressure. The bearing-shell is made in four sections, as is usual in large steam-engine practice, the vertical section being adjusted by means of wedges. The first section of the side-shaft is carried in bearings mounted on the side of the main frame, and provided with ring lubrication. It is driven from the crank-shaft by means of spiral gearing. The second section, which carries

the valve-operating cams, is similarly mounted alongside the cylinders, and on a level with the centre line. It is driven from the first section by means of spur gearing. The distance-piece between the two cylinders, a plain cylindrical barrel, machined and spigoted to the two cylinders, is shown in fig. 152. The upper part is cut away, to permit of inspection of the piston-rod, stuffing-glands, and slipper, and reinforced by steel connecting-stays. The lower part of the distance-piece is machined concentric with the rod, to form a slide for the piston-rod slipper. At the after-end of the second cylinder there is a somewhat similar piece, which also contains a slide for the tail-rod slipper.

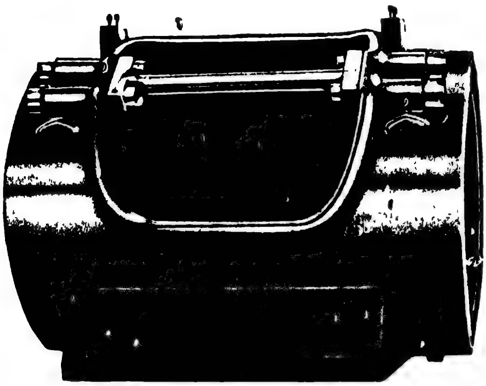


Fig. 152.—Distance-piece

The crankshaft itself calls for no particular comment. It is naturally of very massive construction, and is provided with heavy balance-weights. In this particular engine the shaft is forged solid, and cast-iron balance-weights attached, but the practice of using built-up crankshafts is steadily gaining ground. The connecting-rod is a steel forging, forged in one piece with the big-end bearing, which is afterwards split and lined with white metal. The small end of the connecting-rod is forked, and carries a hardened steel cross-head pin which is rigidly attached to the connecting-rod, and mounted in bearings carried by the cross-head.

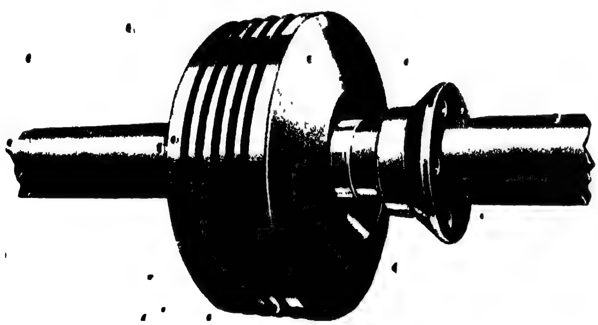


Fig. 153.—Piston with Lo[illegible]ed Nut

The pistons are of cast iron, cast in one piece, and machined all over externally. They are bored out to a good sliding fit on the piston-rod, and are held in place between a conical collar and a conical nut, screwed on to the piston-rod, as shown in fig. 153. A very liberal clearance is allowed between the piston and liner, so that there shall be no actual contact. The piston-rings are of the

ordinary Ramsbottom type, and have no unusual features. The piston body is, of course, water-cooled, the water entering and leaving through the piston-rods. Although Messrs. Erhardt & Sehmer adhere to cast iron for the pistons, many other makers prefer cast steel; and, in some instances, forged mild-steel pistons are employed. As regards unequal expansion, the piston is subject to the same conditions as the cylinder, but it is of a very much more symmetrical form, and, moreover, the rate of heat-flow is less intense than in parts of the cylinder body, so that the problem is by no means so difficult a one. The piston-rod is made in two sections, and is of high tensile steel, machined and ground all over. It is of very liberal dimensions, to ensure against sagging and to resist the severe reversals of stress.

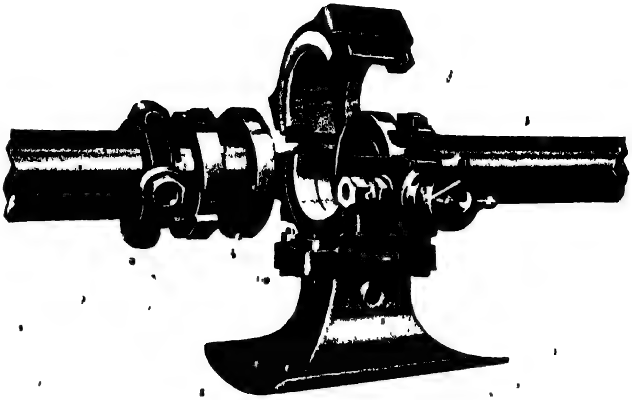


Fig. 154.—Piston-rod Coupling with Cover lifted and Rods separated

The method for coupling the various sections of the piston-rod is shown in detail in figs. 154 and 155. The ends of the rods are screwed and provided with nuts, each of which has a concentric groove turned in it. The slipper, or cross-head, is fitted with a detachable cap, provided with internal flanges which fit into these grooves. The nuts are first screwed on to the ends of the piston-rods and the cap bolted in place. They are then tightened up until the two faced ends of the rod butt firmly together. Thus both the compression and the tension pressures are taken on ample surfaces, and the whole joint is one that can be quickly and easily dismantled.

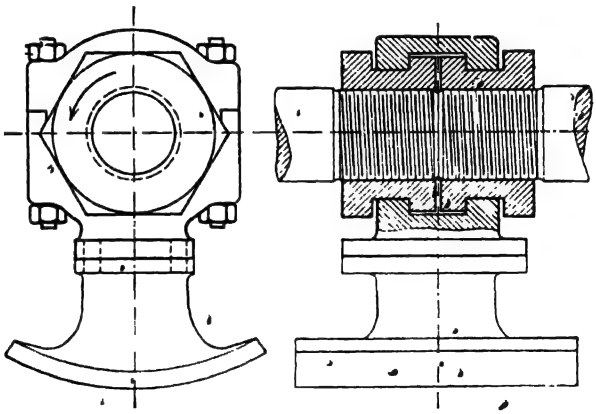


Fig. 155.—Piston-rod Coupling

Water-cooling Arrangements.—For the cooling of the pistons, water is fed under pressure to the middle of the piston-rod, close to the central slipper, by means of walking beams, which consist of a number of links forming a kind of parallel motion. At first sight it would appear that a linkage arrangement such as this, involving a considerable number of oscillating joints, each of which must be packed, would involve a great deal of leakage. In practice, however, very little trouble is experienced, and this system has been found to give much better results than the plain telescopic pipes, owing to the high rubbing velocity through the stuffing-glands in the latter case.

Piston-rod Stuffing-glands.—The stuffing-glands consist simply of a large number of ordinary cast-iron rings which contract on to the rod; but at the extreme outside of each gland a pair of white-metal packing rings is fitted, in very much the same manner as in steam-engine practice. For the sake of convenience, the cast-iron rings are formed into three or four separate groups, each of which is mounted in a separate split housing. The grooves in the housing are of sufficient depth to allow of movement of the rings due to the sagging of the piston-rod. This form of gland, which is used by practically all makers of double-acting engines, is simple and satisfactory. As a general rule, these glands remain tight for fairly long periods, provided that the gas is reasonably clean. It is interesting to inspect the glands of a battery of blowing-engines after, say, three weeks' continuous blowing. Under these severe conditions, the author has generally noted that about 50 per cent give no indication of leakage, another 40 per cent will be blowing perceptibly but not seriously, while from 5 to 10 per cent will be leaking to such an extent that the leakage is audible from some considerable distance, although, expressed in terms of percentage of the cylinder volume, it probably does not amount to anything serious. On the whole, it may fairly be said that although the piston-rod stuffing-glands demand very accurate work, and are somewhat complicated and expensive, they cannot be regarded as a serious source of trouble. Other makers of large double-acting engines employ very much the same form of stuffing-gland; and such variations as there are affect only the small details of construction, and the housing of the rings, but not the general design.

Valves.—The inlet valves used in this engine are of the ordinary conventional design, and carry a separate gas valve mounted on the same spindle, as is usual in horizontal engines. They are operated

direct from a cam on the side-shaft, through the medium of a short rocking lever; but, unlike most large gas-engines, rolling levers are not employed. Messrs. Erhardt & Sehmer explain that they prefer the direct operation from a cam rather than any system of rolling levers and eccentrics, because the latter depend upon very accurate adjustments, which are liable to be upset owing to slight alterations between the relative position of the cylinders and side-shaft, due to expansion.

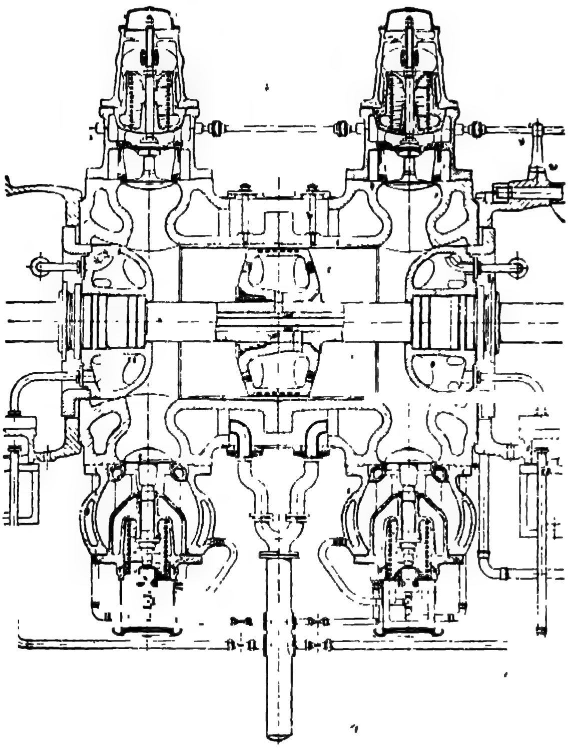


Fig. 156.—Section of Cylinder and Valves

Fig. 157.—Exhaust Valve

The exhaust valves, even in so large an engine as this, are not water-cooled, but very great care is taken to ensure thorough cooling both of the exhaust valve-seating and guide, in order to abstract heat from the head and stem as rapidly as possible. From the cross-section (fig. 156) it will be seen that a small duct is cored round the valve-seating, just below the actual working face, and that water is led up to this duct by means of a small internal pipe. The exhaust valve-seating and cage is an entirely separate piece, which can easily be withdrawn for inspection or cleaning, and has its own independent system of water circulation. The exhaust valve itself is shown in fig. 157. The stem is of steel, and the head of hard cast iron,

screwed and riveted to the stem. It is operated from the same cam, and in precisely the same manner, as the inlet valve.

Governing.—The engine is controlled by throttling both the gas and air. Separate gas and air inlet pipes are led to each valve,

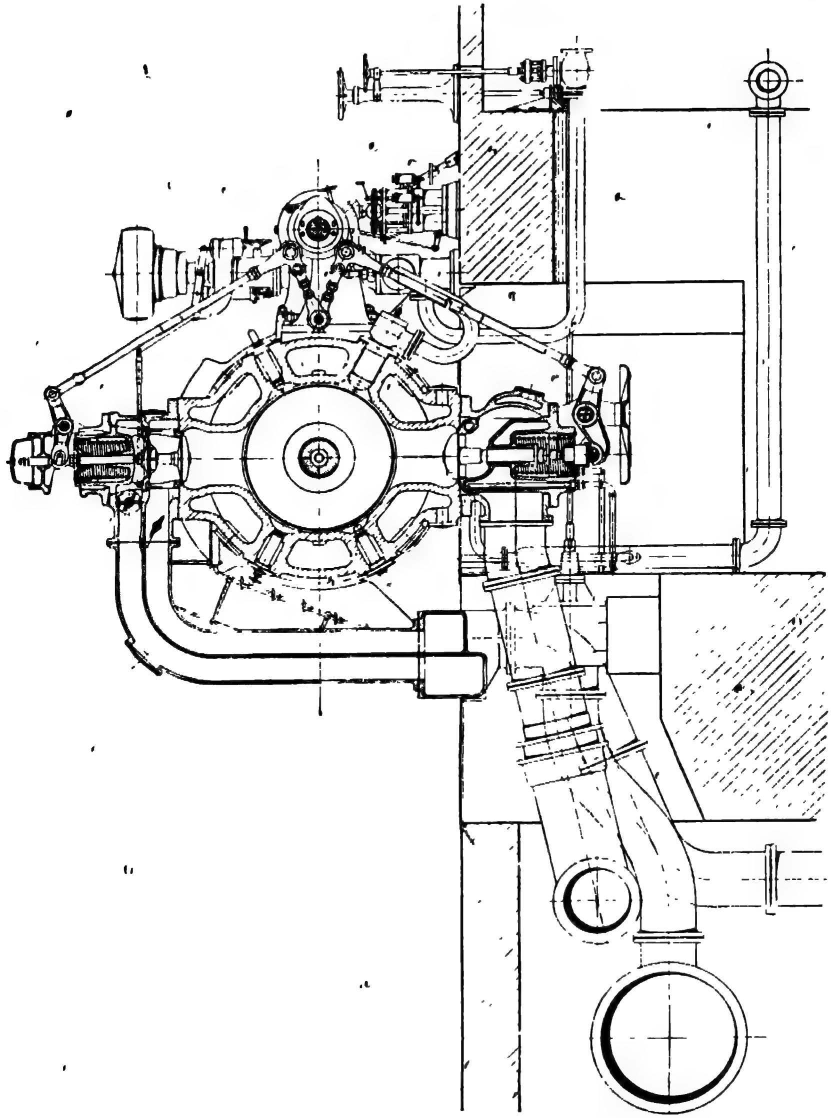


Fig. 158.—Section of Cylinder and Valve Gear

and each individual pipe is fitted with a butterfly throttle valve, all of which are connected together and controlled by the governor, as shown in fig. 158. The relative positions of the butterfly valves are so adjusted that the first movement of the governor cuts down the gas supply, but does not appreciably affect the air supply, and

this is continued until the mixture is at the lowest limit that will ensure regular and reasonably complete combustion. After this point has been reached, the further movement of the governor cuts down the supply of both gas and air in more or less equal proportion. This system of governing is at once simple and efficient, for it provides qualitative governing between the limits in which such governing is possible, in an engine in which there is no stratification, and thereafter the governing is quantitative. Each butterfly valve is capable of independent hand adjustment, so that the mixture can be adjusted for each combustion chamber. Such adjustment is always necessary, on account of the disturbing effect of pulsations in the inlet and exhaust pipes. These effects do not vary in a "constant-speed" engine, and when once the correct adjustment of all the throttle valves has been found, it will remain correct so long as the speed of the engine is not altered, and so long as the heating value and composition of the gas remain uniform.

Ignition.—For ignition, the Lodge high-tension system is employed. This consists of an ordinary high-tension trembler-coil, working in conjunction with a condenser, and gives a very hot spark. Each combustion chamber is fitted with two igniters, connected in series, in order to ignite the gas simultaneously at two points. No attempt is made to place the igniter in the inlet-valve pocket, probably because it is considered unsafe to pierce the cylinder wall at this point, where the internal stresses are necessarily very severe. There can be little doubt, however, that if the igniter were placed in the inlet-valve pocket, qualitative governing could be carried considerably further, and the efficiency of the engine (on medium and light loads) substantially improved.

Efficiency. — Notwithstanding the very large number of double acting engines which are now in service, there are practically no reliable records of fuel consumption available. This is largely to be accounted for by the very great difficulty in measuring large quantities of gas at a low pressure, and when subjected to violent pulsations. From a thermo-dynamic point of view, these large engines are probably not very efficient, because the shape of the combustion chamber is far from favourable, and the area of exposed surface is very large, while the large valve pockets are so placed as to interfere with the free circulation of the gases during combustion, and delay the propagation of the flame. Messrs. Erhardt & Sehmer guarantee that the fuel consumption shall not exceed 8000 B.T.U.s per I.H.P. hour on any gas that the engines are

designed to utilize. This is not a very high figure, and corresponds to an indicated thermal efficiency of only 32 per cent.

The only authentic test, carried out on a double-acting four-cycle engine, that the author has been able to trace, is one made on a small tandem Erhardt & Sehmer engine by Professor Mathot, in 1906, and already referred to in this volume. This engine had two cylinders, each of 24·4-in. bore by 29·52-in. stroke, and developed 600 B.H.P. when running at a speed of 150 R.P.M., with a consumption of 17·8 cu. ft. per horse-power hour of coke-oven gas having a calorific value of 460 B.T.U.s per cubic foot. This corresponds to a brake thermal efficiency of 31 per cent, which is certainly an excellent result. The mechanical efficiency is given as only 83 per cent, which seems much too low a figure for such an engine, in which the piston friction is reduced to a minimum both by the use of an external cross-head and by the high ratio of fluid to inertia pressure. If Professor Mathot's figure of 83 per cent be taken, then the indicated thermal efficiency becomes 37·4 per cent. Since the fuel used contains some 50 to 60 per cent of hydrogen, the compression ratio could hardly be higher than 5·6 : 1, corresponding to an air standard efficiency of 50 per cent. Now, in a four-cycle engine such as this, the relative efficiency could not possibly be higher than about 70 per cent of the air standard, when the unfavourable shape of the combustion chamber is taken into consideration. The indicated thermal efficiency, therefore, could not be higher than 35 per cent, which would give a mechanical efficiency of 88·6 per cent, and this is probably much nearer the actual truth.

In the case of large engines using blast-furnace gas, with which a compression ratio as high as 7 : 1 can be safely employed, it is said that brake thermal efficiencies as high as 33 per cent have been recorded in actual tests. This is, of course, perfectly possible, but the tests referred to are not authoritative and lack confirmation, nor is it easy to see how either the quantity or the heat value of blast-furnace gas could be determined accurately.

In these very large engines, no attempt is usually made to work with high mean pressures on account of the high temperatures involved, and the usual Continental practice is to employ a mean pressure of about 70 lb. per square inch for poor gas, such as producer- and blast-furnace gas, and about 80 to 85 lb. per square inch when using coke-oven gas.

It is obvious that the thorough cleaning of the gas is a consideration of the very first importance. Most waste gases contain

considerable proportions of tar, sulphur, and dust, all of which must be removed, as far as possible, before the reaches the engine. The presence of tar is clearly undesirable, because it deposits in the throttle and inlet valves and other working parts, and soon leads to jamming. Sulphur is not particularly harmful, so long as there are no water-leaks into the cylinder. Dust is obviously harmful, and must be removed as far as it is possible to do so; for not only does it cause excessive wear, but the actual particles of dust are in themselves a fruitful cause of premature ignition.

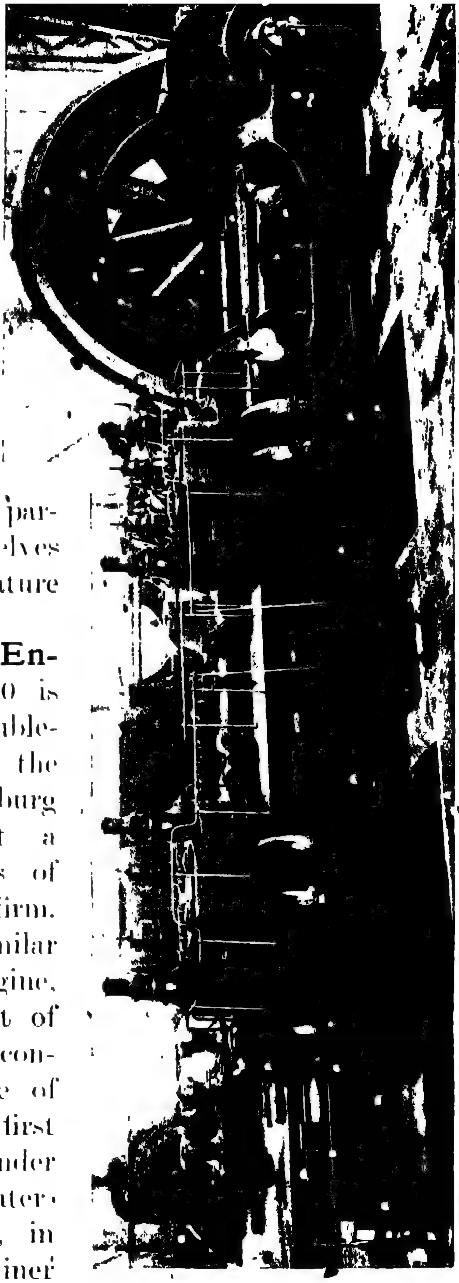


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Fig. 159. —3600-H.P. Twin-tandem Engine

A Large M.A.N. Engine. — In figs. 159 and 160 is illustrated a large tandem double-acting engine, built by the Maschinen - Fabrik Augsburg Nurnberg, who have built a greater number of machines of this type than any other firm. In general design it is similar to the Erhardt - Sehmer engine, but, from a mechanical point of view, there are certain constructional features which are of special interest. In the first place, the whole of the cylinder body, together with its water-jacket, is cast in one piece, in cast iron, and no separate liner is employed. Very great care is taken to ensure against internal stresses due to casting; but it is not at all easy to see how any

provision can be made for the different expansion of the water-jacket and cylinder barrel.

Another feature of interest is to be found in the exhaust valves,

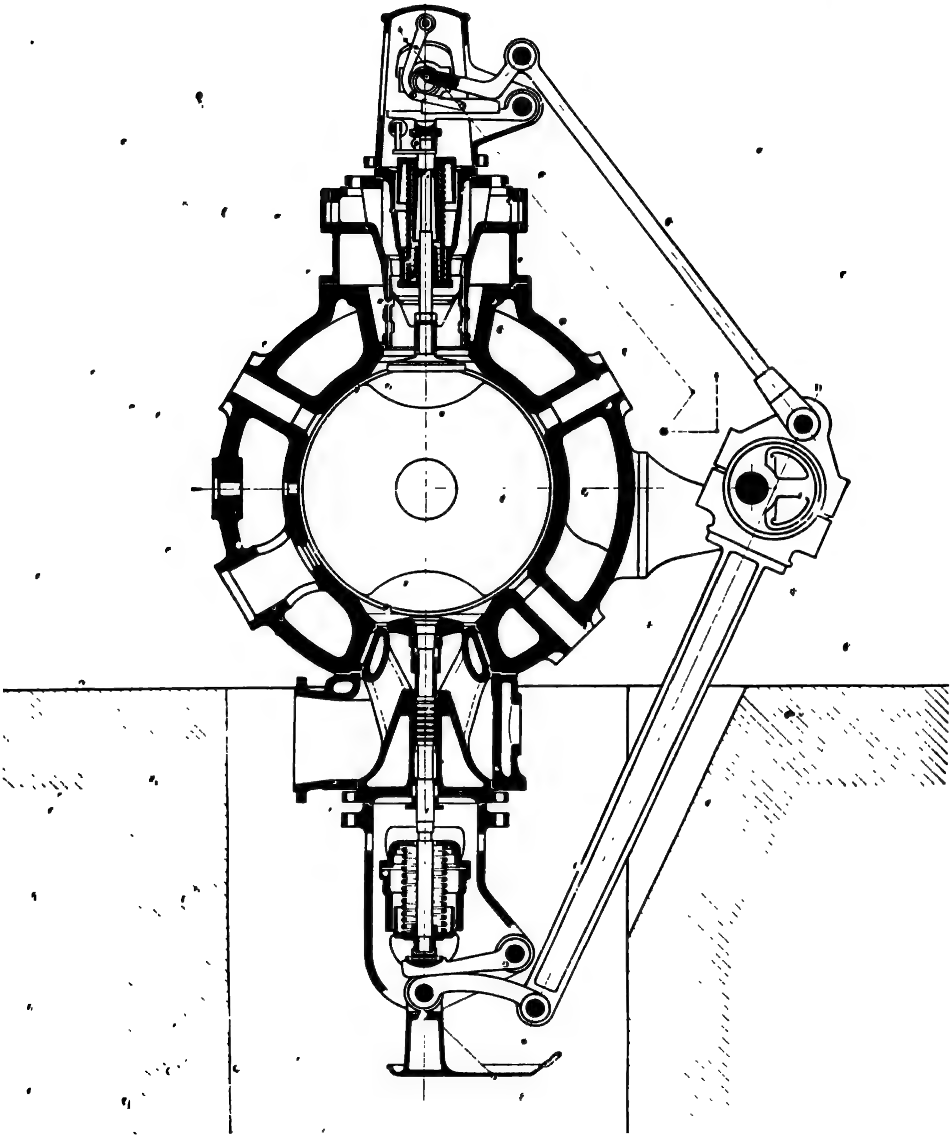
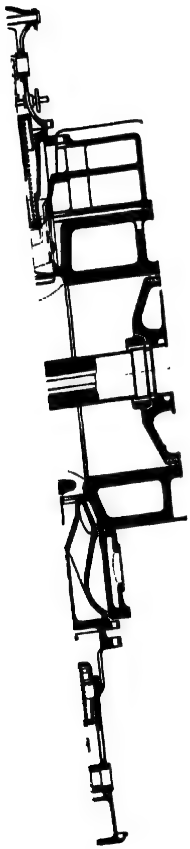


Fig. 161. Nurnberger Gas-engine, showing Valve-operating Gear

which, like the Erhardt-Sehmer valves, are made of steel, with cast-iron heads. The heads in this case, however, are provided with skirts which fit loosely over the water-cooled valve guide, and thus protect the stem from the erosive effect of the blast of high-tem-



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perature gases. Both the inlet and exhaust valves are operated by means of eccentrics mounted on the camshaft through the medium of rolling levers, as shown in fig. 161. The admission of gas is controlled by sleeves mounted on the main inlet-valve stems. These sleeves are provided with a number of ports which register with corresponding ports in a fixed liner, when the valve is opened. In almost all other respects, the Nürnberg engine resembles the Erhardt-Sehmer, and practically all other European designs of large four-cycle, double-acting engines.

The Snow Engine.—In the large gas-engines built by the Snow Steam-Pump Company, of Buffalo, U.S.A., the conventional design has been departed from, and the valves are placed in side pockets, with the inlet arranged vertically over the exhaust valve, as in the National and other vertical engines. The chief advantage of this design is that it makes it possible to place both the inlet valve and the igniter in a pocket, and thus allows of better governing and better efficiency on light loads. On the other hand, the provision of a side pocket, of the depth necessary to accommodate the valves, and the extension of the water-jacket in order to embrace this pocket, must lead to very severe stresses, due to unequal expansion, and there is a considerable danger of cracks developing both in the cylinder body and the water-jacket, at the point where the pocket meets the cylinder barrel.

In the Snow, as in most other American engines, an overhung single crank is employed in place of the more usual double crank as used in Europe. The arguments for and against this practice are, of course, numerous. The American system undoubtedly admits of the use of a very much cheaper crankshaft and connecting-rod. On the other hand, the weight of the main frame is enormously increased, since the stresses are not transmitted directly down it. In the large American gas-engines, the crank disk, balance-weight, and crank-pin are all cast in one piece, in mild steel, and a cast mild-steel connecting-rod is also generally used. This free use of cast steel certainly reduces the cost of the engine, and appears to be entirely satisfactory. In Europe, however, few of the manufacturers have yet acquired sufficient confidence in mild-steel castings to risk their use in such vital parts as the connecting-rods or crankshaft-journals of a large gas-engine. This conservatism may be entirely without justification; but it must also be remembered that, as a general rule, more is expected of a European than of an American engine, and breakdowns are taken more seriously. On the whole,

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Fig. 162.—1200-H.P. Single-cylinder Double-acting Engine driving a Southwark Blowing Engine

the large gas-engine, in America, has been more or less of a disappointment.

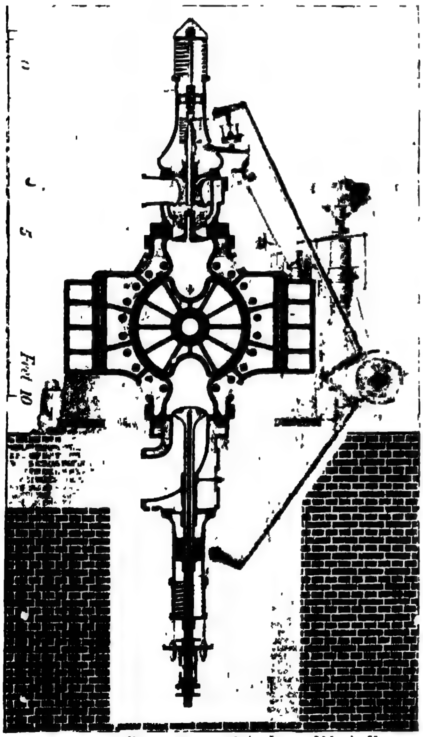

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Fig. 163. Cross-section of Cylinder, showing Valve Gear

Cockerill Double-acting Engine.—The Société John Cockerill, of Seraing (near Liége), Belgium, was one of the first firms to embark on the construction of large gas-engines for blast-furnace gas, and this company created somewhat of a sensation by exhibiting at Paris, as long ago as 1900, a single-cylinder, single-acting, four-cycle gas-engine of 600 B.H.P. This engine had a cylinder bore of no less than 51·2 in., and even to this day no larger cylinder has been constructed. This first engine ran well, and was followed by a large number of others of the same type, many of which are still in successful operation. It was soon realized, however, that the weight and cost of a single-acting engine, of this enormous size, were so great that it could not compete with the double-acting type which was being developed in Germany. The Cockerill Company, however, lost very little time in bringing out a double-acting engine, on somewhat similar lines to the German type.

An example of one of these engines is shown in figs. 162 and 163. This engine, however, has only a single cylinder of 51·2-in. bore, and 55·1-in. stroke, and develops 1200 B.H.P. when running at a speed of 80 R.P.M. As a general rule, two such cylinders are used in tandem, and this is rather an exceptional case. The engine shown is directly connected to a large blowing cylinder, the piston of which is mounted upon a continuation of the main piston-rod. In most respects, the Cockerill double-acting engines conform to what may now be regarded as standard practice, but the construction of the frames is somewhat different. These consist of two long box-girders, which are carried the whole length of the engine, and between which the cylinders are bolted, as shown in the section. In order to permit of free expansion, they are held rigidly by the middle point only.

CHAPTER XXV

THE GOVERNING OF GAS-ENGINES

A good deal has been written from time to time about the governing of gas-engines. The various systems have already been discussed from a theoretical point of view, and it only remains, in this chapter, to consider the practical applications and the mechanical features of these systems. It has already been explained that there are three leading principles of control:—

1. "Hit and miss".
2. Quantitative or throttle governing.
3. Qualitative governing.

In the first principle, the quantity of gas admitted at every firing stroke is always the same, and the speed is controlled by varying the number of power strokes. A separate valve is used for the admission of the gas, operated from the camshaft through the medium of what is generally termed a "pecker", that is to say, a chisel-shaped push-rod, hinged at one end and connected with the governor, so that, when the governor rises, the pecker is lifted clear of the end of the valve and misses it altogether.

In the second principle, the governor controls the quantity of both air and gas in equal proportions, according to the speed.

In the third principle, the governor controls the supply of gas only, the air admission remaining unrestricted. Each of these three systems must be regarded from two points of view: (1) thermodynamic, and (2) mechanical.

Taking first the "hit and miss" principle, and considering the thermodynamic point of view. In an earlier chapter it has been shown that, when running on light loads, the thermal efficiency of the expansion stroke following a series of misses is actually higher than when firing at every cycle, on account of the thorough scavenging and the excess of air present in the cylinder, and that, therefore, the indicated thermal efficiency is greater on light than

on full load, though the difference is comparatively small. This advantage is, however, offset by the very much greater fluid losses during the idle strokes, when running on light loads, with the result that the net indicated thermal efficiency is somewhat lower on light than on full load. In practice, "hit and miss" governing is the simplest and, from a mechanical point of view, the most efficient principle of the three, but it is open to two objections:—

1. The turning moment is very uneven. On light loads there may be only one power stroke in every six or seven cycles, and this involves the necessity for an abnormally heavy fly-wheel, and causes excessive reactionary vibration.

2. The engine is subjected to shocks of equal severity whether it be running on full load or light, and the life of the bearings and other parts is less than when some method of governing is employed which provides an impulse at every cycle.

It is on account of these two objections that the "hit and miss" principle has become obsolete, except for very small engines, and even in that field it is now rapidly dying out.

The mechanical problems connected with the application of "hit and miss" governing are exceedingly simple, for the gear can be so constructed that the load thrown upon the governor is negligible, and the movement required is so exceedingly small that very close governing can be obtained.

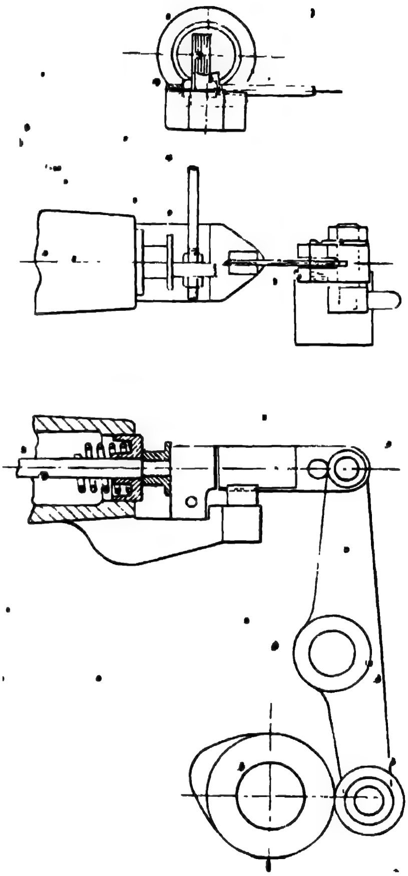

Fig. 164.—"Hit and Miss" Governor

Crossley Governor.—In fig. 164 is illustrated the arrangement adopted by Messrs. Crossley Brothers for their "hit and miss" governed engines. In this case, the gas-valve cap is provided with a wide flat surface, against which rests a small hardened-steel block, suspended from the governor-arm. One side of this block has a series

of V-shaped grooves cut in it, the other side, bearing against the valve cap, being flat. The rocker from the arm carries, at its extremity, a light hardened-steel "pecker", consisting of a thin piece of tool steel, ground down to a chisel edge at one end, and loosely pivoted to the rocker-arm at the other. The lower side of the pecker rests in a V-shaped guide, whose function it is to prevent it swinging sideways, and, at the same time, to allow it a certain amount of lateral play, so that it shall always be able to find its way into one or other of the V-shaped grooves in the governor-block, and not ride on the top of them.

In normal operation, the pecker engages the governor-block at every cycle, forcing it against the flat valve cap, and so opening the gas valve; but, so soon as the load is reduced, and the speed rises, the governor slides the movable block laterally and out of engagement of the pecker, so that the valve is not opened. When running at normal speed, the pecker either engages with the last groove of the block or misses it entirely, the movement required to differentiate between the two being almost infinitesimal. When the engine is stopped, the governor falls so far that the block is slid out of engagement with the pecker in the other direction, and all danger of the engine stopping accidentally with the gas and air valves open is thus eliminated. It is obvious that, with this arrangement, the governor is only called upon to move a very light block through an infinitesimally small distance, and that at a time when it is merely "floating". From the point of view of sensitiveness, it would be impossible to improve upon this system.

Quantitative Governing.—There are three leading methods of quantitative governing in general use:—

1. By throttling the supply of gas and air, either by means of a single throttle valve controlling the mixture, or by means of two throttle valves, one controlling the air and one the gas.

2. By varying the lift of the main inlet valve, which, in this case, is provided with a supplementary valve on the same stem, controlling the gas admission.

3. By varying the cut-off instead of the lift of the main inlet valve, or by using a supplementary cut-off valve.

The first system is commonly employed for engines using town gas, or other gases which are comparatively clean and free from tar. When there is much tar or dirt in the gas, however, throttle valves are liable to become choked and to stick, causing the governor to "hunt", and, in extreme cases, entirely preventing it from function-

ing. The use of a single throttle valve for gas and air involves the addition of a separate mixing valve behind the throttle valve, to determine the proportions of the mixture. Such a valve, as used by Crossley Bros., is illustrated in fig. 124. The arrangement is simple and effective, and the governor is called upon only to operate a light, balanced, butterfly throttle valve, fitted close up to the main inlet valve. The mixing valve takes care of the proportioning of the mixture, and relieves the governor of all but the operation of the single throttle valve. Such an arrangement has also the additional advantage that, in the event of the engine being accidentally stopped, the supply of gas is automatically cut off by the mixing valve. This, of course, is a very important point in the case of small engines, which are often left running unattended.

In some of the larger engines in which throttle governing is employed, two throttle valves are fitted, one controlling the gas and the other the air, and both operated simultaneously by the governor. This arrangement has decided advantages. The butterfly valves can be so adjusted that the first movement of the governor affects the gas valve only, but not the air; this can be continued until the mixture is weakened as far as is consistent with complete combustion, and, thereafter, the two valves operate together.

In this manner the first reduction of the load is affected by qualitative governing, until the limit of this form of governing is reached, and thereafter by quantitative governing over the remainder of the range. The effect of this is, of course, that the maximum efficiency is obtained at the point where the qualitative governing ends, which is generally at about three-quarters of full load, the load on which most engines are generally run in actual service. The use of double valves has also the advantage that it is very easy to vary the proportion of gas and air independently of the governor, or to adjust it separately for the separate cylinders of a multi-cylinder engine, in which the proportions are liable to be seriously affected by the pulsations in the inlet piping. The objection to the arrangement lies in the fact that the governor has two or more valves to operate instead of only one. Whether one or two valves be used, they must necessarily be of considerable size to avoid restricting the supply, and for the same reason their travel must be considerable. This, of course, reduces the sensitiveness of the governing, and necessitates the use of a very powerful governor, if anything like close governing is to be aimed at, unless some form of air or hydraulic relay be employed.

Variable-lift Inlet Valves.—In most of the larger engines using producer or waste gases, in which there is generally a large proportion of tar or dirt, the use of throttle valves is generally undesirable, and the supply of gas and air is controlled by varying the lift of the main inlet valve. For this purpose, an enormous number of different gears have been devised, some quite fantastically complicated. The fundamental idea of most of these gears is the interposition of a lever with a movable fulcrum between the push-rod and the valve. The arrangement adopted on the Crossley engine is shown in fig. 137, and this is typical of modern practice. The swinging fulcrum-rod is suspended above a curved link, and varies the lift of the valve according to the position it takes up. This swinging rod is operated by the governor, and, during the period that the valve is seated, it is out of contact with the curved link, and is floating freely, so that it requires no appreciable effort to swing it to and fro, and has no tendency to react upon the governor. This, of course, is only one of a whole legion of mechanical devices to obtain the same object, but it is a simple and thoroughly effective design. It is somewhat surprising that although there are any number of devices to be found that effect their object without throwing any load upon the governor, yet some engines are fitted with such devices as tapered cams, wedges, &c., many of which require considerable power to operate them, and all of which are, so to speak, unbalanced, and react upon the governor, causing, or at least tending to cause, it to "hunt". The system of varying the period of opening of the main valves, and the employment of auxiliary cut-off valves, have been alluded to. The object of such devices being to reduce the fluid losses during the charging stroke, the mechanical complication introduced by them is generally out of all proportion to the small advantages gained, and now that the design of gas-engines has settled down to more sober lines, they have practically all disappeared.

The use of quantitative governing has become almost universal on gas-engines of medium power, and is very largely used on high-powered engines. It is simple, and, since it provides for one impulse at every cycle, it has the great merit of giving an even turning moment, good balance, and freedom from shock.

Qualitative Governing.—In an earlier part of this volume, it has been shown that if it were possible to control the speed and power of an engine by varying the density, but not the quantity, of the working fluid, the efficiency would increase progressively as the

load was reduced. There seems to be every reason for supposing that, at the point of no-heat supply, the actual efficiency would be equal to the air standard efficiency, as is shown in the curves, fig. 13, Vol. I, and that in an ordinary engine, with a 6 : 1 compression ratio, the efficiency would probably increase from about 35 per cent at full load to about 47 per cent when running light. Unfortunately, however, if the gas supply be reduced beyond a certain point, combustion will be first retarded and incomplete, and finally will not take place at all.

The actual range over which reasonably complete combustion can be relied upon is very small. In the case of town gas, the range of density available is from about 90 down to about 50 or 55 B.T.U.s per cubic foot. At the higher density the temperatures are very high, and the stresses very severe. Moreover, owing to the excessive heat loss, and the loss due to the increased specific heat of the gases, the efficiency is low. For practical purposes, the highest mixture-density that can be used with advantage is generally about 75 to 80 B.T.U.s per cubic foot, and if the lowest be taken as 50 B.T.U.s per cubic foot, it follows that the range of load over which qualitative governing can be employed is only about 35 per cent, which is not nearly sufficient.

Although many engines now on the market are provided with qualitative governing, yet these engines are either intended to run under conditions which afford a practically uniform load, or they are provided with some supplementary form of governing, such as "hit and miss" or throttle governing, on the lighter loads. In practice, a qualitative-governed engine cannot as a rule be relied upon to run regularly over a range of load exceeding 50 per cent, unless provided with some such supplementary system. If the range be carried too far, and the mixture weakened beyond a certain point, combustion becomes so incomplete and slow that it is still continuing even at the time when the inlet valve opens, with the result that the entering charge is ignited, causing a back-fire through the inlet valve.

In almost all the systems of qualitative governing, as applied to four-cycle engines, provision is made for the admission of air alone, during the first portion of the suction stroke, followed by gas and air during the latter portion, the proportion of gas and air admitted during this latter portion of the stroke remaining approximately the same at all loads, but the quantity is varied by the governor. The object of this, of course, is that the contents

of the cylinder shall consist of nearly pure air next to the piston, and of combustible mixture next to the igniter. In practice, there is no doubt that some such stratification does occur, though to a very limited extent, for engines governed by this method have a range of power which certainly exceeds the range of inflammability of the mixture. The loss, however, due to retarded and incomplete combustion counterbalances the gain that might be expected from the higher efficiency theoretically obtainable, with the net result that qualitative governing, as at present applied, is little or no more efficient than quantitative, and is very decidedly more sensitive and tiresome to look after.

In large double-acting engines, a mechanically-operated auxiliary gas valve is generally fitted, either concentric with or alongside the main inlet valve. The timing of this valve is controlled by the governor, through the medium of one of the many forms of trip-gear, so that it is opened earlier or later in the stroke, according to the load, but is always closed either just before, or simultaneously with, the main inlet valve.

In the design of the trip-gears there is, as might be expected, a great deal of variety; but any gear that will wear well, is tolerably silent in operation, and throws no serious load upon the governor, is suitable. In some few cases, designers have abandoned all attempts at obtaining stratification, and have fallen back on a plain throttle valve in the gas pipe only. With such a crude arrangement, it is obvious that the range over which the engine will run is equal to, but not greater than, the range of inflammability of the mixture, i.e., in the case of town gas, only about 35 per cent.

Crossley Qualitative Governor.—About the year 1907, Messrs. Crossley Brothers built some large tandem single-acting engines, developing from 500 to 600 B.H.P., which were provided with qualitative governing. For these engines, a peculiarly ingenious inlet valve and governor device was adopted, which, from the successful manner in which it worked, deserves careful consideration. The device is illustrated in section in fig. 165. The main inlet valve is operated directly from the inlet cam, and its operation is unaffected by the load or speed. The gas valve is carried concentrically on the stem of the main valve. This valve is free to move in one direction, but is compelled to close when, or slightly before, the main valve closes, by means of a small collar screwed to the main valve stem. Below this collar, a second collar is also attached to the main valve stem, and a spring is fitted between it

and the gas valve. By this means, the gas valve is forced open by the compression of the spring, when the main valve is opened, and is closed positively by the screwed collar when the main valve closes.

The upper portion of the gas valve is formed into a piston which is a close, but free fit, in a short cylinder formed in the top of the valve cage. This cylinder is completely closed, except for a small hole through which air can enter, and the area of which is controlled by a small piston valve connected to the governor. If this valve be

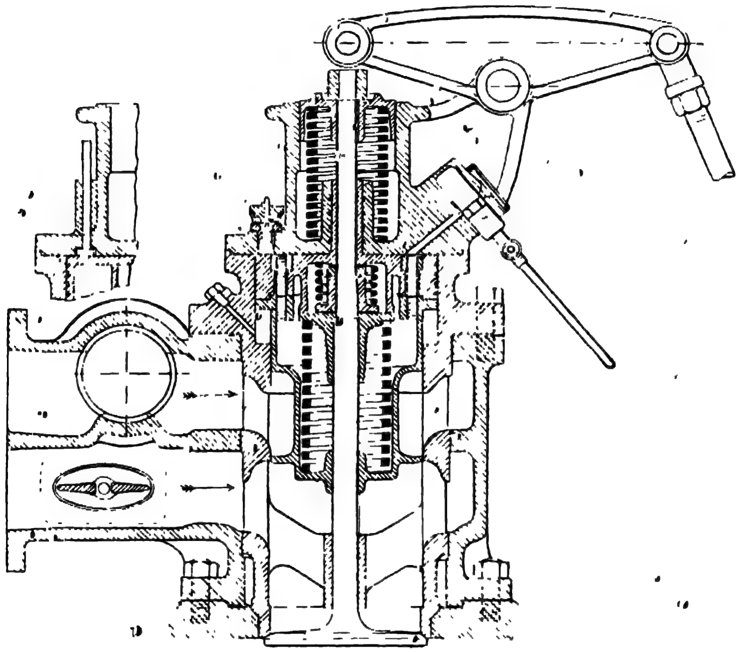

Fig. 165.—Governing Gear (Crossley)

in such a position that the air inlet is completely cut off, it is obvious that the gas valve cannot open without forming a vacuum in the cylinder. The spring opening the gas valve, however, is not of sufficient strength to produce a vacuum in the cylinder, hence, under these conditions, it will not open at all. If now the small piston valve be shifted by the governor to such a position as shown in the illustration (fig. 165), air can enter the cylinder slowly, and the gas valve will open as soon as sufficient air has entered to relieve the vacuum. That is to say, it will open some time after the main inlet valve, and will remain open until it is closed by the closing of the main valve. It is obvious that with this arrangement the opening of the gas valve can be delayed to any desired extent by merely moving a small and light balanced piston valve through a very

small range. All that the governor is required to do is to vary the position of this minute valve.

The gas valve, in this case, is in the form of a balanced piston valve, and has a lap of about $\frac{1}{4}$ in., so that it has to travel through this distance before opening the gas inlet port. In order to close the gas valve completely, the collar on the main valve stem is supplemented by a small buffer spring, which ensures the return of the piston and gas valve to the top of its stroke, and which expels any air which may have leaked into the vacuum cylinder, through the small check valve shown in the cover. The whole arrangement is simple and highly ingenious, the time of opening of the gas valve is under the most perfect control, and that without throwing any perceptible load on the governor, or requiring anything but a very trifling movement.

Tests carried out on this engine by Professor Nicholson showed that the variations of speed, when the load was instantaneously dropped from 600 to 50 horse-power, did not exceed 1·66 per cent. No figures are given as to the efficiency of the engine on the lighter loads, but indicator diagrams, taken with a mean pressure of 55·6 lb. per square inch, i.e. about 60 per cent full load, show regular, though retarded, combustion. An illustration of this engine, which shows very clearly the actual arrangement of governing, is shown in fig. 166.

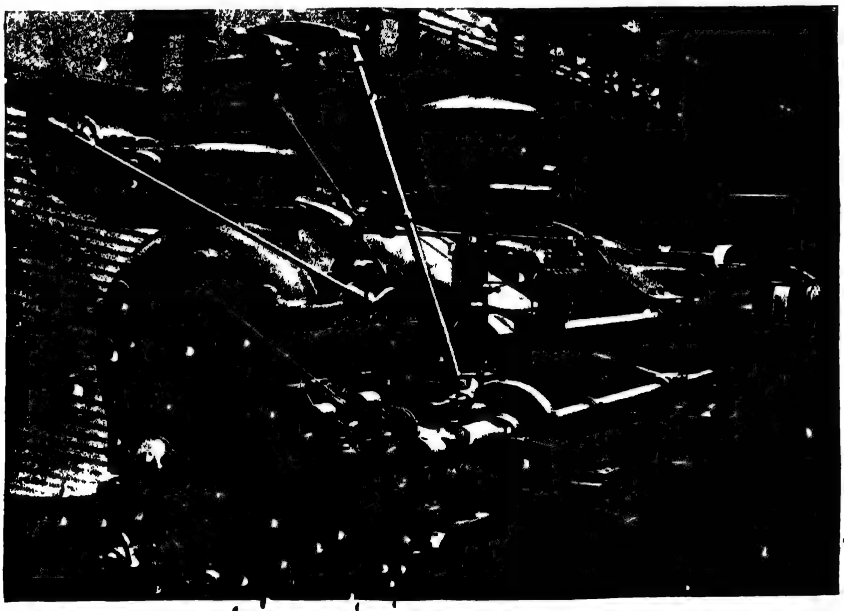

Fig. 166.—Governing Gear on Tandem Engine of 500 to 600 H.P. (Crossley)

CHAPTER XXVI

VAPORIZING OIL-ENGINES

Very soon after the success of the ordinary gas-engine had been thoroughly established, a demand arose for, and manufacturers turned their attention to, the development of a modified form of gas-engine, which could run on ordinary paraffin oil, and so enable the engine to be used in districts where town gas (then the only fuel for gas-engines) was unobtainable.

The engine produced in response to this demand followed the accepted gas-engine design, and differed only in this respect, that provision was made for vaporizing the oil, and a lower compression ratio was provided. Such engines were termed oil-engines, and, for a long time, represented the only form of engine consuming liquid fuel on the market. With the advent of the Diesel oil-engine, and still later of the so-called semi-Diesel engine, both of which are designed throughout for the express purpose of consuming liquid fuel, and which are capable of consuming practically any form of this fuel, even down to the heaviest residues, the field of usefulness of the modified gas-engine has been very much restricted. At the same time, the development of gas-producers, even in comparatively small powers, thus enabling gas-engines to be used in places where ordinary town gas is unobtainable, has still further narrowed down the scope of the vaporizing oil-engine. At the present moment, there appears to be every prospect that the ordinary vaporizing type of oil-engine, using refined paraffin, will be driven from the market, or at least restricted to a very limited field, by the severe competition from the producer gas-engine on the one hand, and the semi-Diesel engine on the other, either of which can produce the same power for about one quarter of the fuel bill.

In very small powers, up to 5 or 6 horse-power, the paraffin oil-engine still finds a considerable market for such purposes as agricultural work and the electric lighting of country houses; but, in

these small sizes, a new competitor arises in the form of the petrol-engine, which, although it uses a more expensive fuel, is cleaner, handier, and more reliable than the paraffin engine. In the somewhat specialized field of marine work, the vaporizing oil-engine is still extensively used, even in powers up to 200 horse-power; but this is probably because its chief competitor, the semi-Diesel engine, with few exceptions, has not yet been developed for marine work on rational lines.

Vaporization.—In the ordinary paraffin engine, the oil is vaporized either within or without the cylinder, and the vapour treated exactly as the gas is treated in a gas-engine, that is to say, it is mixed with air, compressed, and ignited in the usual way. The problem would be simplicity itself if it were possible to convert paraffin into a gas by merely warming it. Unfortunately, commercial paraffin is not a definite chemical compound with a definite boiling-point, but is a collection of all those hydrocarbons which distil over between 200° F. and about 600° F., and includes the whole range from Nonane, C_9H_{20}, to Pentadecane, $C_{15}H_{32}$, and frequently even higher up the range. The lower end of the range is limited by the restrictions imposed as to the flash-point, but the upper end is limited only by the commercial honesty of the distiller and the tendency for the still heavier fractions to discolour the fuel, and so disclose their presence. Since the boiling-point of the various constituents which go to form commercial paraffin varies from 200° F. to 600° F., it is clear that, in order to vaporize the whole of the fuel, its temperature must be raised to at least 600° F., otherwise fractional distillation will take place, and the heavier fractions will collect in the vaporizer.

Now, if paraffin be heated to a temperature of 600° F., some of the lighter fractions will be "cracked", that is to say, they will be split up into carbon and hydrogen, the former being deposited in the vaporizer, which would very soon become choked. It is evident, therefore, that it is not possible merely to boil the paraffin and treat the vapour as a fixed gas, as can be done with petrol. If, however, the fuel be heated in a vaporizer, through which air is passing, at a temperature of about 500° F., "cracking" will be prevented, and nearly the whole of the fuel will be vaporized: but, if the temperature falls much below 500° F., only partial vaporization will occur, and, if much above, it is liable to crack. Now if the whole of the fuel and the air supplied to the engine were raised to a temperature of 500° F. before entering the cylinder, it is obvious that

the volumetric efficiency would be reduced to little more than half of that which would be obtained with a normal charge. Also, owing to the high suction temperature, only an exceedingly low compression ratio could be employed, for the temperature would very soon reach a point at which severe detonation and pre-ignition would occur.

From the above considerations, it is evident that it is not practically possible merely to vaporize the fuel and then treat it as a fixed gas. The other alternative is to pulverize the oil to a very fine degree, generally by passing air through it at an extremely high velocity, and, as far as possible, to maintain it in this finely atomized condition until it is ignited. Unfortunately, the finely divided particles of fuel can only be kept in suspension so long as the air is travelling at an exceedingly high velocity. So soon as this velocity falls, as, for instance, after passing the inlet valve and entering the cylinder, the particles immediately show a tendency to coalesce and precipitate on the walls of the cylinder, or upon any portion of the induction pipe where a change of velocity or direction occurs. Any fuel that is so precipitated escapes combustion, because it cannot be surrounded with the necessary supply of air. A small portion of the fuel so precipitated will burn slowly from the surface, leaving a heavy carbon deposit, but the greater proportion will pass through the engine completely unburnt.

In practice, most of the engines which employ separate external vaporizers rely partly on heating the fuel and air, and partly upon pulverization, and a fairly satisfactory, or at least a workable, compromise is arrived at.

When quantitative governing is relied upon, as in the case of marine oil-engines, the amount of precipitation increases as the load is reduced, owing to the reduction in the velocity of the air, so that, when running on light loads, the proportion of fuel that escapes unburnt is very large, and carbonization of the cylinder walls and pistons takes place with alarming rapidity. Owing to the very narrow range over which a mixture of paraffin and air is inflammable, qualitative governing is out of the question; hence it follows that the vaporizing type of oil-engine is virtually driven to rely on some form or other of "hit and miss" governing. It is true that, in the marine oil-engines, quantitative governing is almost invariably employed, but these engines are practically always run at or near their full load, and are only throttled down for very brief periods. If run for any length of time at a much reduced load, they will soon carbonize up and give trouble.

In all vaporizing oil-engines, owing partly to the pre-heating of the working fluid, and partly to the tendency of hydrocarbons of the paraffin series to detonate, only a very low compression ratio can be employed, and hence the efficiency is abnormally low, as compared with a gas-engine. The ratio usually adopted is about 3·5 : 1, giving an air standard efficiency of approximately 40 per cent. The limit of compression is, of course, set by the temperature and pressure at which a mixture of paraffin vapour and air will ignite spontaneously.

Carburation by pulverization alone, and without any addition of heat, has scarcely—so far—been found practical, for the fuel cannot be so finely pulverized in a cold condition that it will not precipitate almost immediately on leaving the spraying chamber. It might be possible in the case of a very high-speed engine, but, in this case, there would not be time for the finely-divided particles to burn completely, so that what might be gained by the use of a higher compression would be lost through retarded and incomplete combustion. In general practice, it is usual to rely as much as possible upon pulverization, and to pre-heat the charge as sparingly as possible, and thus effect a kind of compromise, in which the fuel is partially vaporized and partially pulverized. In such cases, the mean suction temperature is kept down to about 250° F., allowing of a compression ratio of about 3·5 : 1 without serious risk of pre-ignition or unduly rapid combustion. It has already been explained in an earlier volume that the compression ratio can be raised, and the temperature kept down, by the admission of water to the cylinder, in a finely-divided state. The water, during the compression stroke, is evaporated and converted into steam, a certain proportion of the heat of compression being absorbed in overcoming the latent heat of conversion. The presence of steam in the working fluid reduces the efficiency slightly, owing to its high specific heat, but the gain in efficiency (due to the higher compression ratio) more than compensates for the loss on this score. The use of water is, however, to be avoided if possible, for, if the whole of the water be not completely evaporated, it may combine with the sulphur, which is always present in the fuel, to form a corrosive acid, and, in any case, it is exceedingly detrimental to the piston lubrication.

Internal Vaporization.—The foregoing remarks refer primarily to those engines which are merely standard gas-engines, with the addition of a vaporizer or pulverizer outside the cylinder. There is, however, another class of vaporizing oil-engine in which

the fuel is admitted to the cylinder in the liquid form, and is vaporized after admission. In this case, the cylinder is provided with an uncooled head, or bulb, to which the oil is fed and in which it vaporizes. This bulb is generally separated from the rest of the cylinder by means of a restricted neck; and, in some cases, is completely isolated by a valve fitted between the bulb and the cylinder proper. The capacity of the bulb is such that it cannot contain sufficient air for the ignition of the fuel. Cold air is drawn into the cylinder through the inlet valve, in the usual manner, and this air, during the compression stroke, is driven into the bulb, where it meets, and mixes with, the vaporized oil. At the end of the compression stroke, the bulb contains a mixture of vapour and air in the correct proportion, and this mixture is ignited either by electric ignition or by the heat of compression. In the latter case, which is the more usual, the compression ratio must be so adjusted that the ignition temperature and pressure are not reached until the end of the stroke, for the range of combustible mixture on the rich side is not so narrow that the contents of the bulb cannot be ignited a considerable period before the end of the compression stroke.

This type of oil-engine has the advantage over the external vaporizing type in that the air is not heated before admission, and that, therefore, a larger charge can be taken into the cylinder, and a higher mean pressure carried. If the bulb be completely isolated from the remainder of the cylinder, by means of a timing valve, and communication established only when the compression stroke has been completed, then it is clear that any reasonable compression ratio can be safely employed; but it is not easy to see how a valve which has to pass the working fluid at the time of its maximum pressure and temperature, can be made to last and be kept tight.

There is yet another system which seems to hold out a good deal of promise, namely, vaporization under a pressure considerably less than atmospheric. Even with a comparatively small reduction of pressure, the temperature at which the heavier fractions of paraffin will vaporize is substantially reduced. In a four-cycle engine it is, of course, a simple matter so to arrange and time the valve gear that the vaporizer is kept under a partial vacuum. So far as the author is aware, there is not on the market, at the present moment, any oil-engine in which this system is adopted; but he understands that Messrs. Broom & Wade, of High Wycombe, at one period constructed a number of engines using this principle, for tractors, and obtained very satisfactory results.

From all the foregoing considerations, it is obvious that whatever be the type of vaporizing oil-engine employed, it is always a matter of vital necessity to maintain the vaporizer at an approximately uniform temperature, in order to avoid "cracking" on the one hand and partial vaporization on the other. In those engines in which a vaporizer is employed, the necessary heat may be supplied either from a lamp or from the exhaust from the engine. By the use of a lamp, it is possible to maintain the vaporizer at practically any desired temperature, irrespective of the load on the engine, for the loss of heat in overcoming the latent heat of the fuel is trifling in comparison with the loss by radiation, and the heat taken up by the air passing through. Hence the variations in the quantity of oil passing through the vaporizer with different loads does not affect its temperature to any appreciable extent. The use of a separate lamp, however, adds considerably to the complication of the engine, and is, naturally, not in favour, especially since paraffin blow-lamps are none too reliable at the best of times.

The alternative method is to heat the vaporizer by the exhaust gases from the cylinder, but here the difficulty arises that the temperature of the gases varies according to the load, and especially is this the case in engines governing on the "hit and miss" principle. This difficulty is generally met by the provision of a bye-pass valve, which can be adjusted by hand, and which allows a certain proportion of the products of combustion to pass direct to the silencer, without travelling round the vaporizer, when the engine is running on full load. When lightly loaded, the bye-pass valve is closed by hand, and the whole of the exhaust gases are compelled to pass through or round the vaporizer. If the vaporizer be very heavily constructed, so that it has considerable thermal storage capacity, a tolerably even temperature can be maintained.

In those engines in which vaporization takes place within the cylinder, or in the inlet-valve chamber, and in which the necessary heat for vaporization is provided by the temperature of some uncooled portion of the cylinder or valve chamber, the problem of maintaining an even temperature is still more difficult, for the thermal storage capacity in this case is reduced by the cylinder-jacket circulating-water with which the parts must be more or less in contact. In this case there is little doubt that the best method is to control the engine by means of the exhaust valve, so that, when running on light loads, the governor holds the exhaust valve open during the suction stroke, and hot exhaust gases are drawn into the cylinder

in place of cold air. Alternatively the valve may be kept closed throughout the cycle, so that the exhaust gases are alternately compressed and expanded. In either case the temperature of the jacket can be maintained tolerably uniform.

This, of course, is simply a modified form of "hit and miss" governing, in which the governor acts on the exhaust instead of on the gas valve, as in a gas-engine. In order to prevent a charge of cold air being drawn into the cylinder as well as the exhaust gases, the air inlet valve is usually automatic in its operation, and is loaded by a sufficiently powerful spring to prevent it from opening during the suction strokes, when the exhaust valve is open. This method undoubtedly answers admirably, for engines governed in this manner are generally able to deal with varying loads without any attention to bye-pass valves, &c., such as is generally required when exhaust-heated vaporizers are used. Moreover, the comparatively clean running and clear exhaust, at all loads, indicates that a fairly even temperature is being maintained.

Efficiency.—As might be expected from the limited compression ratio, the thermal efficiency of vaporizing oil-engines is very low. The air standard efficiency is generally only about 40 per cent, and the high suction temperature, necessarily involving high temperatures throughout the whole cycle, increases both the heat loss and the losses due to the increasing specific heat of the working fluid at high temperatures, with the result that the relative efficiency seldom exceeds about 60 per cent, corresponding to an indicated thermal efficiency of 24 per cent, assuming perfect and complete combustion, which is very seldom obtained. The brake thermal efficiency also is very low, because—

1. The mean pressure is low owing to the low efficiency and the loss of volumetric efficiency due to pre-heating of the charge; and

2. The precipitation and incomplete combustion, that invariably occur to a greater or lesser degree, result in gumming of the piston-rings and impaired piston lubrication.

3. In order thoroughly to atomize the fuel, it is necessary that the velocity of the incoming air shall be increased considerably, as compared with usual practice, and this results both in a loss of volumetric efficiency and a substantial increase in fluid pumping during the suction stroke.

All these conditions react upon the mechanical efficiency, which seldom exceeds about 80 per cent, bringing the brake thermal

efficiency down to about 19 per cent, as against 25 per cent that might be obtained with a gas- or petrol-engine with the same compression ratio. In practice, a brake thermal efficiency of even 19 per cent is seldom if ever obtained from a vaporizing oil-engine, and 16 per cent is a much more usual figure.

The Ruston-Proctor Engine. Among the recent designs of vaporizing oil-engines, one of the most interesting is the small engine recently brought out by Messrs. Ruston-Proctor, in response to the demand for a small, reliable, and cheap engine, and which is shown in figs. 167 and 168.

The normal power is rated at 5 brake horse-power, when running

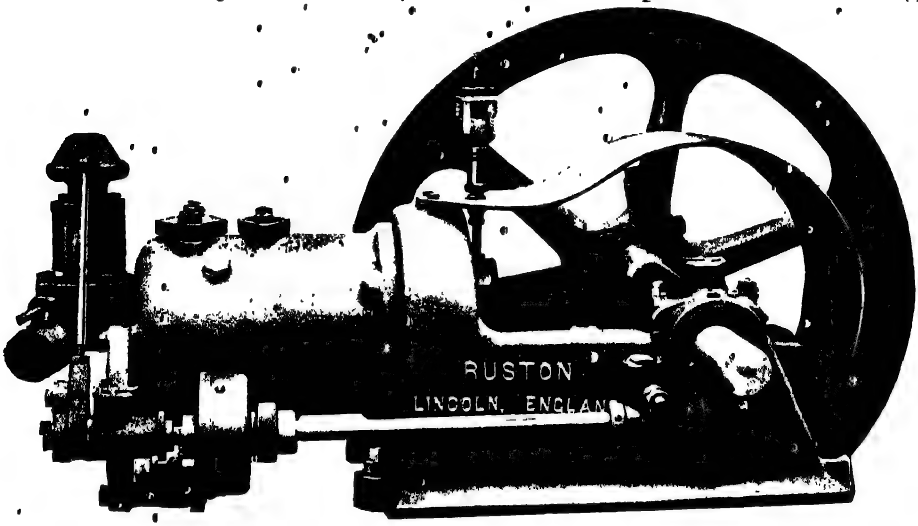

Fig. 167.—The Ruston-Proctor Vaporizing Oil-engine

at a speed of 360 R.P.M.; the bore and stroke of the cylinder are $6\frac{1}{2}$ in. and 9 in. respectively, and the normal power rating is based on a brake mean pressure of 51·7 lb. per square inch. In general design, it follows accepted gas-engine practice as applied to small engines, but the interesting features lie in the arrangements for the admission, pulverization, and vaporization of the oil. On the back of the cylinder-head a small bulb or pocket is fitted. This bulb is unjacketed, and is maintained at a sufficiently high temperature to ensure vaporization, or, at least, to prevent precipitation of the oil, by the heat of combustion. In the side of the bulb is fitted an ignition-tube of large capacity, which really forms an extension of the bulb. The metal of this tube extends to a considerable distance from the water-jacket, and can only get rid of its heat by radiation.

After each combustion stroke the ignition-tube is left full of

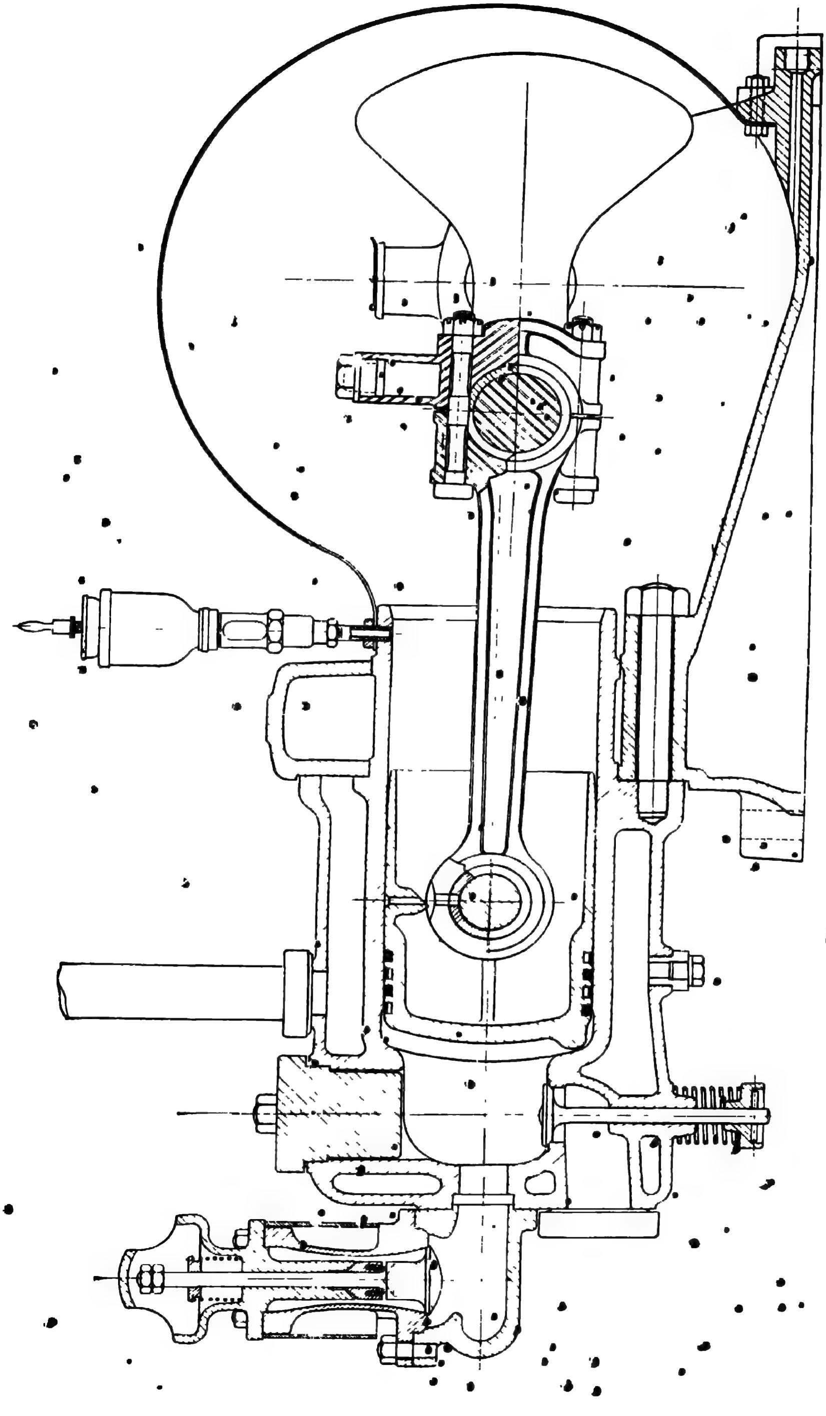

Fig. 168.—Sectional Arrangement of Ruston-Proctor Engine

inert exhaust gases, and, therefore, although its temperature is high, it cannot ignite the charge in the cylinder until the end of the compression, when the combustible mixture has been driven back into it and reaches a point at which the temperature is sufficient to cause ignition. It is obvious that to ensure ignition taking place at the right moment, the temperature gradient down the tube must be carefully regulated. Since the tube can only get rid of its heat by radiation, it follows that the temperature can be regulated by controlling the amount of radiation, and this is accomplished by means of an adjustable, asbestos-lined sleeve, which can be slid along the ignition-tube, and thus prevent the loss of heat from a greater or lesser portion of it.

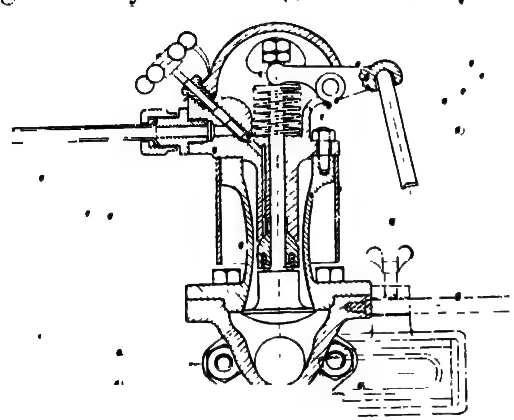

Fig. 169.—Ruston-Proctor Engine. Inlet Valve

The arrangements for the admission of the fuel are particularly interesting. The main inlet valve is operated by means of a rocking lever from the inlet cam. This lever, however, does not bear upon the valve stem, but upon the inlet-valve spring, so that, during the suction stroke, the valve is not mechanically opened, but is relieved of all spring pressure. Thus it can open automatically, unhindered by any resistance due to the spring, but is closed at the correct period, so soon as the spring is released by the cam. The cage surrounding the inlet valve is made in the form of a Venturi tube, so that the air entering near the top shall pass down between the cage and the inlet-valve guide at an exceedingly high velocity. The admission of oil is controlled by a second small coned valve, mounted loosely on the main inlet-valve stem, and held up against its seat by means of a light spring, as is usual in gas-engine practice. This valve is operated by means of a small collar pinned to the valve stem, so that it is opened when the main valve is opened.

The object of the subsidiary spring is merely to ensure that the fuel-controlling valve is seated when the main valve is closed. The fuel is admitted through a needle valve, shown at the side of the main valve in fig. 169, and passes down through a small hole drilled in the main-valve guide to the fuel-valve seating. The operation is

as follows: When the main inlet valve is opened by the suction in the cylinder, air is drawn down through the Venturi at a high velocity; at the same time, the fuel valve is opened, admitting oil through the small hole in the valve guide. The oil then passes over the conical fuel valve, and on meeting with the air at the point of maximum velocity, is very finely pulverized. The oil and air, the former in a finely atomized but not vaporized condition, then enter the uncooled bulb in the cylinder-head. Owing to the high temperature of the walls of this bulb no precipitation can occur, and the particles of oil are held in suspension and partially vaporized. The vaporization continues throughout the compression stroke, and is probably more or less complete by the end of it. It will be observed that in this engine—

1. The air is not heated before admission to the cylinder.
2. No attempt is made to vaporize the oil before admission to the cylinder.
3. Every effort is made to pulverize the oil as finely as possible, by drawing the air past the oil inlet valve at the highest possible velocity.
4. Precipitation of the finely atomized particles of oil is prevented by maintaining the air at a very high velocity, and avoiding any abrupt changes of velocity or direction until it has entered the cylinder.
5. Precipitation of the particles of oil within the cylinder is avoided as far as possible by preventing them from coming into contact with cooled surfaces until after they have been at least partially vaporized. No doubt a certain amount of precipitation does occur on the walls of the cylinder barrel and on the water-cooled surface of the combustion chamber, but it has been avoided as far as it is possible to do so.

It is clear that, in such an engine as this, a compromise must be arrived at; for if the area of uncooled surface be increased, the degree of precipitation will be reduced. On the other hand, the suction temperature will be raised, involving the use of a lower compression ratio, to avoid pre-ignition, with reduced efficiency. The exact area, and shape, of uncooled surface which will give the best compromise between these two conflicting conditions can only be found by trial and error, and there is no doubt that Messrs. Ruston-Proctor have satisfied themselves upon this point. The compression pressure employed in this engine is 55 lb. per square

inch, corresponding to a compression ratio of about 3·53 : 1, and giving an air standard efficiency of 40 per cent. The fuel consumption is given as 0·73 lb. per B.H.P. hour, corresponding to a brake thermal efficiency of 18·2 per cent, which is an excellent result for so small an engine of this type.[1] The leading dimensions of this engine are as follows:—

Bore	6·5 in.
Stroke	9 in.
Number of cylinders	1.
Piston area	33·2 sq. in.
Swept volume	298·8 cu. in.
Compression ratio	3·53 : 1.
Maximum B.H.P.	7.
R.P.M.	360.
Piston speed	540 ft. per minute.
Brake mean pressure (mp)	51·7 lb. per square inch.
Diameter of inlet valve	2·25 in.
Lift of inlet valve	0·34 in.
Effective area of opening	2·4 sq. in.
Diameter of exhaust valve	1·875 in.
Lift of exhaust valve	0·5 in.
Effective area of opening	2·76 sq. in
Ratio, piston area to inlet-valve area	13·8 . 1.
Weight of piston	25 lb.
Weight of connecting-rod	28 lb.
Weight of reciprocating parts	36 lb.
Weight of reciprocating parts per square inch of piston area	1·185 lb.

The brake thermal efficiency of this engine is 18·2 per cent, and the mechanical efficiency is estimated by the makers to be 78 per cent, giving an indicated thermal efficiency of $\frac{100}{78} \times 18{\cdot}2 = 23{\cdot}3$ per cent, and a relative efficiency of about 58 per cent.

Qualitative governing, that is to say, governing by controlling the quantity of fuel at each stroke, leaving the air unrestricted, is out of the question in such an engine as this, where there is no provision for stratification, and is, in any case, especially difficult because the range of mixture of paraffin vapour and air, over which complete combustion can take place, is particularly narrow. Moreover, this system of governing, even if it were feasible, would not solve the problem of maintaining a uniform temperature.

The method adopted in this and in many other such engines,

[1] The lower effective calorific value of paraffin is taken as 19,150 B.T.U.s per pound.

is to control the speed, either by holding the exhaust valve open or closed. Of the two, it is, perhaps, preferable to hold it closed rather than open. The highly heated exhaust gases are then alternately compressed and expanded in the cylinder, and thus serve to maintain the temperature approximately uniform. It is obvious that, during this period, the inlet valve must remain closed, or air and fuel will be drawn into the cylinder as well as exhaust gases. As a general rule, the inlet valve is not mechanically operated, but is fitted with a light spring, of sufficient strength to prevent the valve from opening when not required. This method, however, is open to two objections:—

1. The action of the valve is very noisy, automatic spring-loaded valves having a tendency to flutter, thus setting up a very disagreeable noise.

2. In order to ensure the valve closing at the correct time the spring must be fairly stiff, and this causes considerable wire-drawing, and loss of volumetric efficiency.

In the Ruston-Proctor engine the valve is operated partly mechanically and partly automatically, as already explained. A stiff spring is used, and the spring tension relieved during the suction stroke, to obviate wire-drawing, but released again mechanically at the correct moment to ensure rapid closing. This method provides the advantages of the mechanically operated valve, and at the same time allows the valve to remain seated while the engine is governing.

The Campbell Engine.—The Campbell Gas-engine Company, of Halifax, have for many years been constructing vaporizing oil-engines, not only in small but also in quite large sizes. That there can be any future for large engines of this type seems unlikely in face of the competition from the much more efficient Diesel and semi-Diesel engines. The large vaporizing oil-engines were, no doubt, designed and introduced at a time when the semi-Diesel engine was unknown, and their manufacture will, in the author's opinion, be discontinued as soon as the public have gained sufficient confidence in the high-compression type.

The Campbell engine is built in sizes up to 70 B.H.P. per cylinder, and the following data have been taken from a twin-cylinder engine developing 140 B.H.P. when running at a speed of 180 R.P.M. In general principle the operation of the Campbell engine is similar to that of the Ruston-Proctor. In both cases the

engines are governed on the exhaust valve, and the fuel is admitted immediately behind the main inlet valve, without any pre-heating of the air, to an uncooled chamber forming an extension of the combustion chamber. In the Campbell engine, however, the main inlet valve is entirely automatic in its action; also, there is no separate fuel valve, but the fuel is led to a number of small holes drilled in the seating of the main inlet valve. By this means the entering air is drawn at a high velocity past the oil-admission holes, thoroughly pulverizing the oil. In effect, the two systems are similar, but the Ruston-Proctor system of using a small supplementary valve to

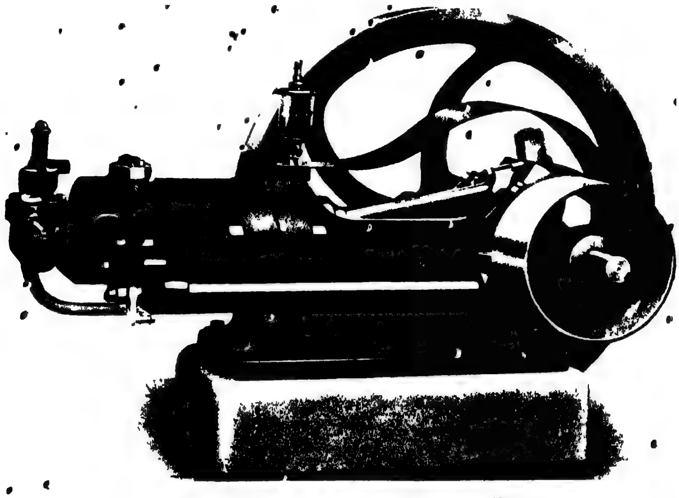

Fig. 170.—Crossley Farm Oil-engine, small size

control the oil admission appeals to the author as being the more mechanical arrangement of the two. In all other respects the Campbell vaporizing oil-engines resemble the gas and semi-Diesel engines made by this firm.

Crossley Oil-engine. Messrs. Crossley Brothers, of Openshaw, Manchester, have for many years been constructing vaporizing oil-engines up to quite large powers. During recent years, however, the success of the semi-Diesel type built by this firm has led to the abandonment of the vaporizing type, except for small powers. They are now made in two distinct models, the design used for powers up to $5\frac{1}{2}$ B.H.P. having an overhung cylinder and no liner, as in the Ruston-Proctor engine. From $5\frac{1}{2}$ B.H.P. upwards, all the engines are built with the cylinder-jacket integral with the main frames, and with a separate cylinder liner and combustion-head, as in the usual

gas-engine practice. This construction is, of course, more expensive than the overhung cylinder, but it provides a far more rigid structure, and certainly makes for smooth and silent running.

In fig. 170 is shown a photograph of the smaller type, and in fig. 171 of the larger. In so far as the admission and vaporization of the fuel are concerned both types are identical, and both resemble the Ruston-Proctor in all essential features. In fig. 172 is shown a section through the vaporizer and admission valve, from which it will be seen that it differs but little from the Ruston-Proctor

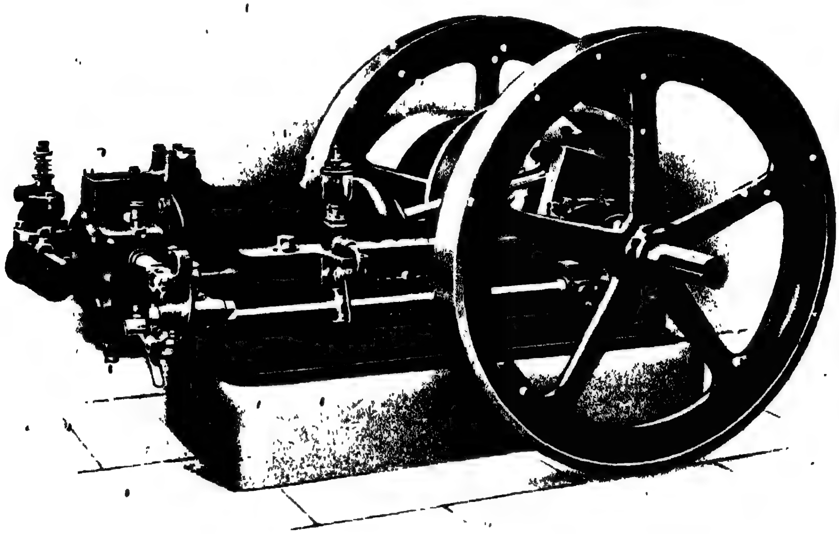

Fig. 171. Crossley Farm Oil-engine, large size

arrangement. In both cases the oil is admitted at a point very near the main inlet-valve port; and, in both cases also, the fuel inlet valve is a separate piece fitted loosely over the main valve stem, and operated by a collar on that stem. A small portion of the combustion chamber is left unjacketed, and the same type of ignition-tube is employed.

On closer examination, however, it will be noticed that one very important feature in the Ruston-Proctor engine is absent in this case, namely, the careful shaping of the inlet passage to avoid any change of direction or velocity. In the Crossley engine, the passage for the air is restricted around the fuel-admission valve, but this restriction is purely local, and opens out into a chamber of considerable capacity immediately above the inlet-valve heads. Consequently, the air will pass the fuel valve at a high velocity; but, immediately

after, the velocity will be reduced, and then suddenly accelerated again through the main valve. This, in the author's opinion, may cause violent eddying, and may tend to precipitation of the particles of the oil on the valve chamber, unless, as is probably the case, the temperature of the chamber (at this point) is too high for such precipitation to occur. In any case, eddying would permit the

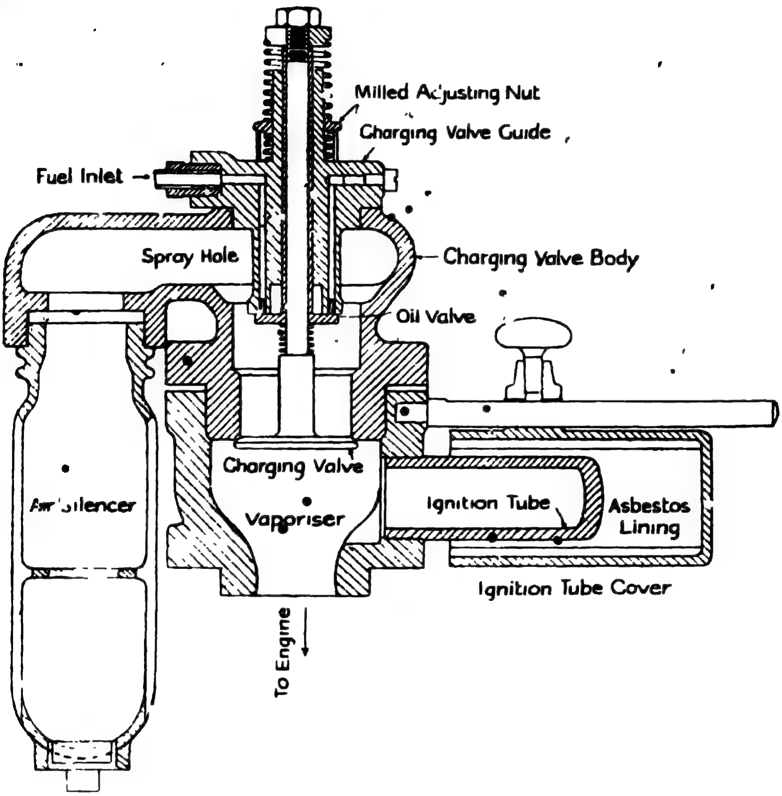

Fig. 172.—Sectional View of Vaporizer and Charging Valve, Crossley Engine

finely-divided particles of paraffin to coalesce again, and must be somewhat detrimental to the atomization.

As in the Campbell oil-engine, the main inlet valve is operated automatically, and is therefore provided with a light spring. This spring can be adjusted by means of a milled nut, shown in section, and the lift of both the main and fuel valves controlled thereby. The engine is governed on the exhaust valve by means of an inertia governor. The valve is operated through the medium of a pecker-rod, which either hits or misses it, so that when the speed is in excess the pecker misses altogether and the valve remains closed, the exhaust gases being retained in the cylinder, and alternately

compressed and expanded until the speed falls and normal operation is resumed. The governor gearing is shown in detail in fig. 173, from which it will be seen that, in practice, two pecker-rods are employed, one of which operates direct on the valve spindle and the other on a spring-loaded block attached to the spindle. The pecker-rod operating the valve has a square end, but that operating the block is ground down to a chisel edge. The block itself is fitted with a V-groove, and, above the groove, is tapered off slightly; also, the clearance between the block and the lower chisel-edge pecker is greater than that between the valve and its pecker.

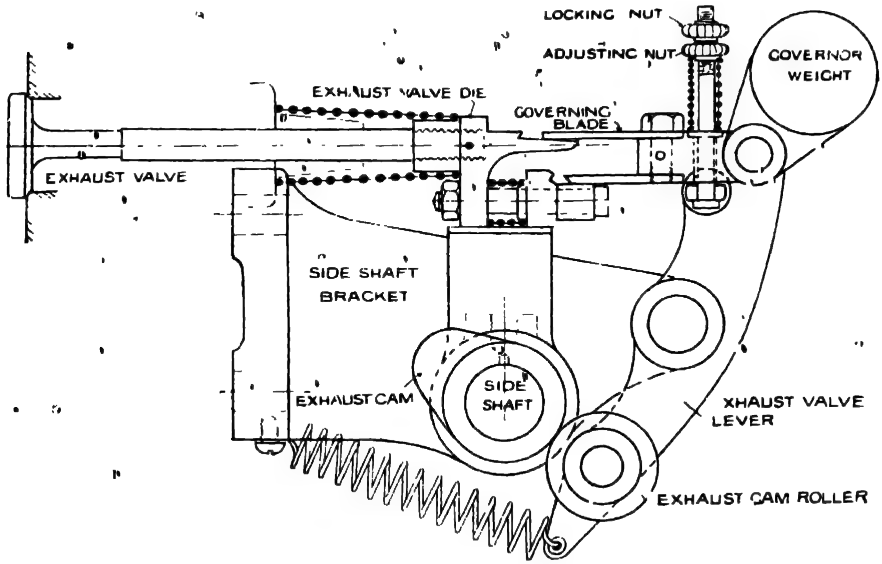

Fig. 173.—Crossley Engine Governor

The object of all this is to prevent what is generally known as "nibbling", that is to say, to prevent the pecker from partially engaging with the valve stem and then slipping off. The two pecker-rods are connected together with a distance-piece between them, and it is obvious that, owing to the lesser clearance, the chisel-edged pecker comes into engagement with the block first, and either drops into the V-groove or slides over the top of it. In doing so it ensures that the upper pecker, which is constrained to move parallel with it, shall either come into fair engagement with the valve stem or miss it altogether. Partial engagement is thus prevented. All this is, of course, a detail refinement, but it is upon such details that the success of an engine largely depends, and this one is well worth noting.

The leading dimensions of the 8-B.H.P. Crossley oil-engine are as follows:—

Bore	6 in.
Stroke	11 in.
Number of cylinders	1.
Piston area	28·26 sq. in.
Swept volume	311 cu. in.
Maximum B.H.P.	8.
R.P.M.	340.
Piston speed	624 ft. per minute.
Brake mean pressure (mp)	60 lb. per square inch.

The brake mean pressure, 60 lb. per square inch, is somewhat high for an engine of this class, and this is doubtless due to the small amount of uncooled surface, and therefore the low suction temperature, as compared with most other oil-engines of the same type. No data are available from which any estimate of the mechanical efficiency can be obtained, nor do the makers state the compression ratio or pressure employed, so that it is not possible to investigate the working of the engine. The following figures are stated by the makers to be the best test results obtained, and are about 10 per cent better than they are prepared to guarantee:—

Load.	Fuel consumption (lb. per B.H.P. hour).	Brake Thermal Efficiency.
8	0·88	15·0 per cent.
6	1·03	12·9 ,,
4	1·34	10·0 ,,

The large drop in the brake thermal efficiency between full and half load suggests that the mechanical efficiency is unusually low. In order to make the indicated thermal efficiency constant at all loads, which is what, at first sight, it would appear to be, with this system of governing, the mechanical efficiency on full load would have to be less than 70 per cent, which is very improbable. Considering the low piston speed and high mean pressures, there seems little reason why the mechanical efficiency should be less than 80 per cent. A possible explanation is that it is at least 80 per cent, and that the indicated efficiency falls with the load, because, while the engine is governing, and the exhaust gases are alternately compressed and expanded, they are losing heat, and at the same time there is a certain loss by leakage. Thus, at the end of each expan-

sion stroke the inlet valve opens slightly and a certain amount of fuel is taken into the cylinder, but in such small quantities that it is excessively diluted with exhaust products, and cannot be utilized. This, of course, is pure conjecture, but it seems very probable that something of this kind would occur, and it would certainly explain the rapid drop in the brake thermal efficiency. If this is really the case it could probably be obviated by the simple expedient of increasing the amount of lost motion between the main and fuel inlet valves, so that the former could open slightly without opening the fuel valve, and without loss of fuel.

The Gardner Engine.—Messrs. L. Gardner & Sons, of Patricroft, Manchester, have for many years been building vaporizing oil-engines for both marine and stationary work. The marine engines are all of the vertical, enclosed high-speed type. The stationary engines are of the horizontal type, and are similar to the gas-engines built by the same firm.

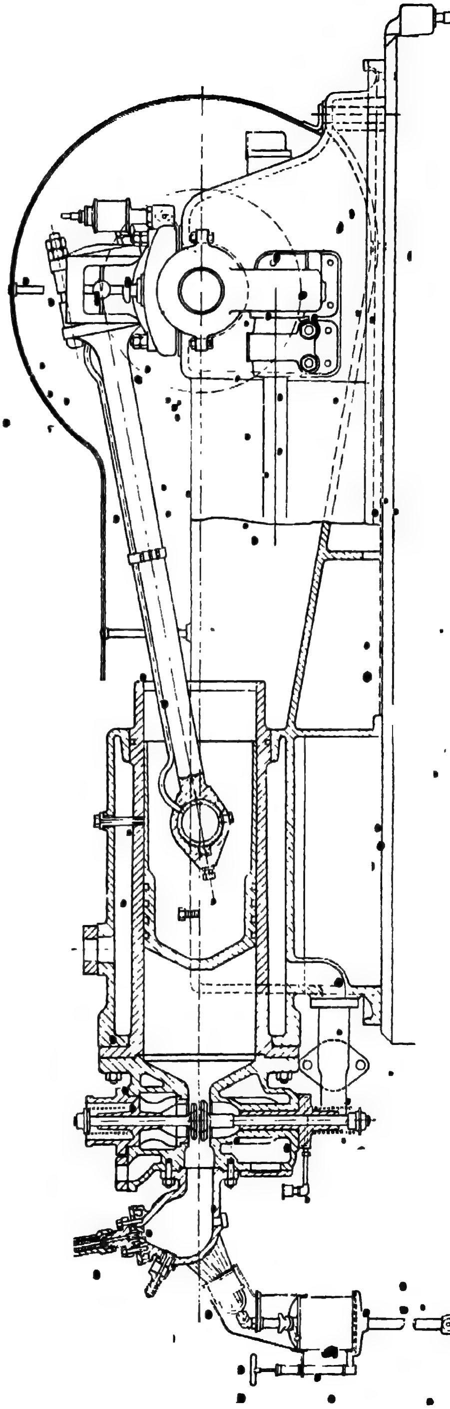

Fig. 174.—Gardner Oil-engine

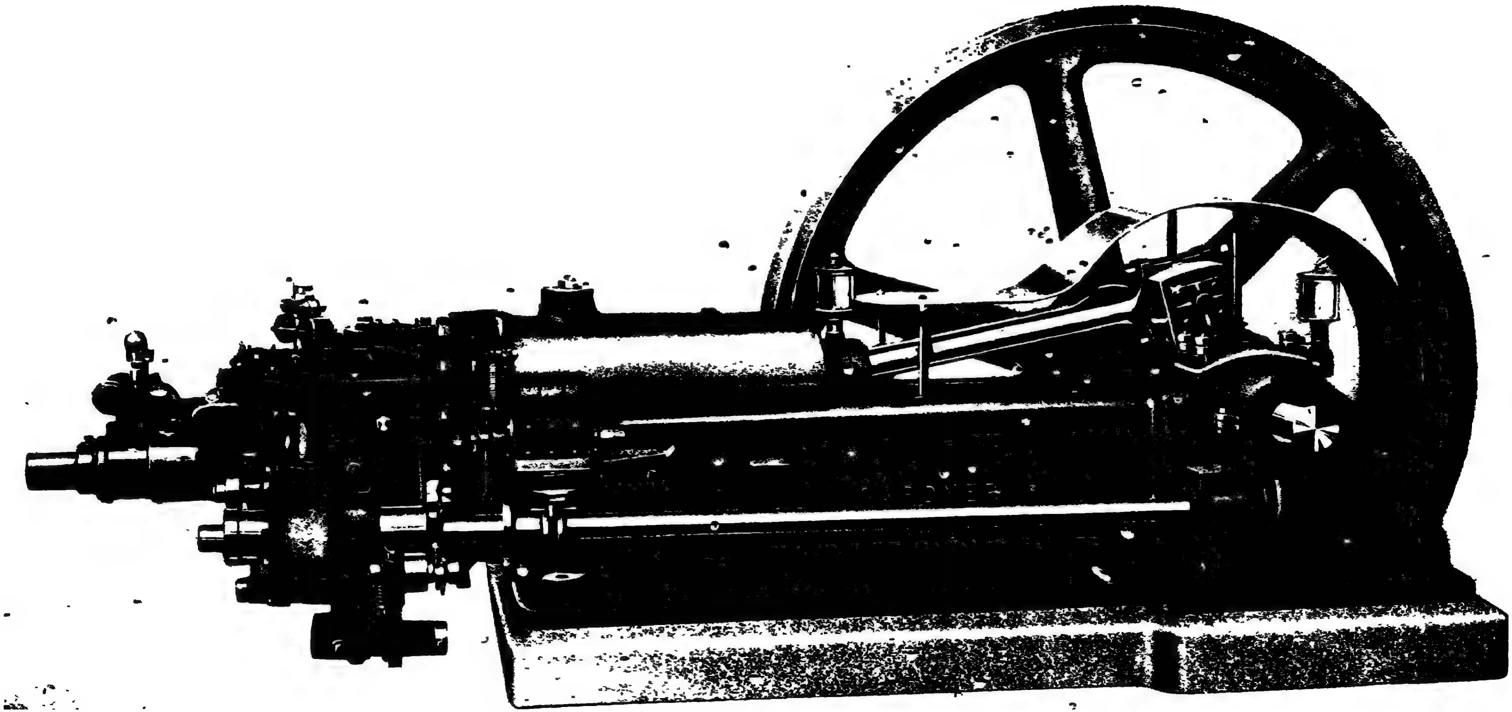

Fig. 175. 18-B.H.P. Gardner Oil-engine

The Gardner oil-engines differ from those already described in several important respects:—

1. The bulk of the air enters the cylinder direct through the main inlet valve, without being mixed with the fuel, and without coming into contact with any uncooled part, so that it is not heated previous to compression. Consequently, a greater weight of air can be taken in per cycle, and the volumetric efficiency is not interfered with.

2. This engine does not govern on the exhaust, but relies upon

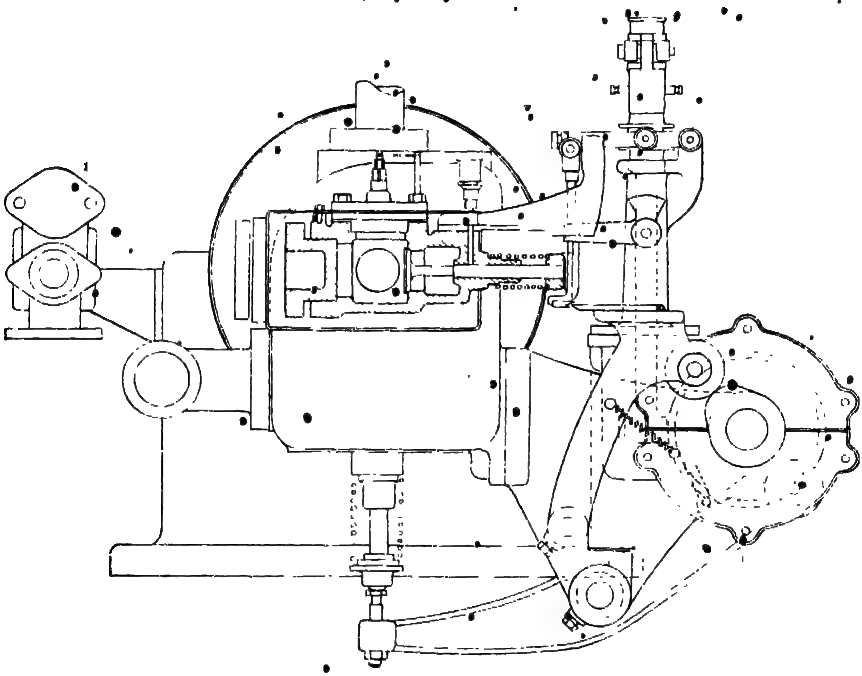

Fig. 176.—Gardner Engine. Fuel Feed and Governor Mechanism

"hit and miss" governing on the fuel only. Consequently, means have to be provided for maintaining the temperature of the uncooled parts when running on light loads. For this purpose a separate lamp is employed, and kept burning during the whole time that the engine is running, unless the load is nearly full, and quite constant, in which case the lamp may be dispensed with.

In fig. 174 are shown sectional drawings of the 18-B.H.P. Gardner oil-engine, and in fig. 175 a photograph of the same. Fig. 176 is a sectional drawing showing the oil-feed and vaporizing arrangements. The oil is fed to the vaporizer by means of a small plunger-pump, whose primary function it is to measure out exactly

the correct proportion of oil for each cycle. The leading dimensions of the 18-B.H.P. Gardner oil-engine are as follows:—

Bore	8·5 in.
Stroke	16 in.
Number of cylinders	1.
Piston area	57 sq. in.
Swept volume	912 cu. in.
Compression ratio	3·86 : 1.
Maximum B.H.P.	18.
R.P.M.	240.
Piston speed	640 ft. per minute.
Brake mean pressure (ηp)	65 lb. per square inch
Diameter of air-inlet valve port	2 in.
Lift of air-inlet valve	0·5 in.
Effective area of opening	3·14 sq. in.
Diameter of fuel-inlet valve port	1 in.
Lift of fuel-inlet valve	0·25 in.
Effective area of opening	0·785 sq. in.
Diameter of exhaust-valve port	2·5 in.
Lift of exhaust valve	0·625 in.
Effective area of opening	4·92 sq. in.
Ratio, piston area to combined inlet area ...	14·5 : 1.
Weight of piston	47 lb.
Weight of connecting-rod	87 lb.
Weight of reciprocating parts	76 lb.
Weight of reciprocating parts per square inch of piston	1·33 lb.

The most striking feature of the above figures is the high brake mean pressure of 65 lb. per square inch, which is an exceptionally good result for such an engine, and is due, no doubt, to the fact that the bulk of the air escapes heating until after the end of the suction stroke.

The following results are given by the makers as the best figures for fuel consumption and efficiency that they have obtained on a test of one of these engines:—

Load (B.H.P.).	Fuel Consumption (lb. per B.H.P. hour).	Brake Thermal Efficiency.
18	0·705	18·8 per cent.
13·5	0·750	17·7 ,,
9	0·838	15·8 ,,
4·5	1·142	11·7 ,,
0	2·25 (total per hour)	—

These results are superior to any that have been obtained from

the engines previously described, but that is exactly what would be expected from the higher mean pressure and compression ratios that can be employed. The mechanical efficiency of this engine is not stated, but by calculation the figure arrived at is 83 per cent. If this figure be taken, then the results become:—

B.H.P.	I.H.P.	Mechanical Efficiency.	Indicated Thermal Efficiency.
		Per cent.	Per cent
18	21·7	83	22·7
13·5	17·2	78·5	22·5
9	12·7	71	22·2
4·5	8·2	55	21·3
0	3·7	—	21·7

With an engine such as this, using "hit and miss" governing on the fuel, the indicated thermal efficiency might be expected to remain nearly constant at all loads, and it is evident that the figure taken for the mechanical efficiency cannot be very wide of the mark. The air-standard efficiency is approximately 42 per cent, and the relative efficiency 56·2 per cent. The actual brake thermal efficiency of this engine is fairly high, but it is obtained at the expense of a considerable amount of extra complication as compared with the Crossley, Ruston-Proctor, and Campbell engines. Also, the temperature of the vaporizing surfaces is not automatically maintained, but requires the use of external heating, which must be applied judiciously. But in this connection it must be admitted that Messrs. Gardner have succeeded in producing a remarkably satisfactory and reliable paraffin-burner, which requires the minimum of attention. Again, the higher compression ratio employed necessitates the use of water injection on full load, which is not a very desirable feature. It is, of course, a matter which the makers alone can decide as to whether the extra complication involved in such a system is justified by the higher efficiency or not.

Gardner oil-engines have long since established for themselves an excellent reputation for reliability and efficiency, thanks to their excellent workmanship and the careful and conscientious testing which every engine undergoes.

Hornsby-Ackroyd Engine. In the Hornsby-Ackroyd oil-engine, and many others of the same type, a somewhat different principle is employed. In these engines the combustion chamber is

divided into two compartments, separated from one another by a narrow restricted neck. The outer compartment is in the form of a plain bulb, which is either partially or entirely unjacketed, while the other compartment is completely jacketed and contains both the air-inlet and exhaust valves. The cycle of operations is approximately as follows.

Beginning with the suction stroke, cold air alone is drawn in through the main inlet valve. At the same time paraffin in a liquid form is sprayed, by means of a force-pump, into the heated bulb, which is left full of residual exhaust products. During the suction stroke the paraffin is vaporized by the heat from the bulb and the hot residual exhaust gases, but, owing to the restricted neck, it does not come into contact with the air in the cylinder. At the end of the suction stroke the cylinder proper is filled with nearly pure air, and the bulb with paraffin vapour and exhaust gases. During the compression stroke the air in the cylinder is driven through the restricted neck into the hot bulb, where it mixes with the paraffin vapour. During the greater portion of the compression stroke, and until quite near the end, the quantity of air that has entered the bulb is insufficient to permit of combustion, and it is only during the last portion of the compression stroke that a combustible mixture is formed. At the extreme end of the compression stroke the contents of the bulb are in the correct proportion for combustion, and ignition takes place partly from the heat and pressure of compression and partly from the heat of the bulb.

It would appear that in this system the actual time of ignition would depend upon very delicate proportioning, not only of the size of the bulb, but also of the exact quantity of oil admitted, and the exact temperature of the walls. Fortunately, however, paraffin —with all its vices as a fuel—has one good feature, namely, that the temperature of ignition is largely dependent upon the pressure of compression, so that even though the temperature of the contents of the bulb remained constant, the point of ignition might be determined by the compression pressure alone. The degree of compression does, of course, require careful adjustment, and is dependent upon the exact grade of fuel which the engine is using; but neither the uncooled surface of the bulb nor the quantity of the fuel require a degree of accuracy of adjustment that is not easily obtainable in a commercial engine. For the adjustment of the compression an opening is provided in the

water-jacketed portion of the combustion space into which plugs of various lengths can be fitted, until the requisite compression ratio is obtained.

A very curious feature of the Hornsby, and other engines of this class, is that it is possible to vary the loads over a very considerable range by qualitative governing. This is no doubt due in a great measure to the stratification which the hot bulb provides, and it is probable that the same engine, run as a gas-engine, could be made to govern on the qualitative system over a wide range of load. With an oil-engine such as this, however, it is very surprising that qualitative governing should be possible, because:

1. At light loads the temperature of the bulb will fall owing to the lower temperatures; and since the heat of the bulb is largely relied upon to ignite the charge, it is reasonable to suppose either that it will not ignite it on light loads at all or that, on full load, it will light prematurely.

2. If the charge of oil admitted to the bulb at each cycle is varied, it is clear that the point at which sufficient air has entered to enable combustion to take place will also vary.

Presumably these two conditions to some extent counteract one another; so that on light loads, although the bulb is colder, the mixture contains sufficient air for combustion at an earlier period in the stroke and probably does in fact start to burn earlier, but owing to the further addition of relatively cool air and the presence of a greater proportion of exhaust products in relation to the fuel, combustion is delayed and the rise of pressure comparatively slow. It is usual for prolonged running on light load to apply external heat to the bulb. The actual governing of the engine is effected simply by allowing a certain proportion of the oil from the oil-pump to pass back into the fuel tank.

Little data appears to be available as to the actual results obtained from Hornsby engines, although a very large number of such engines have been built, and are in successful operation. Tests carried out on a Hornsby oil-engine by the Royal Agricultural Society at Cambridge in 1894 yielded the following results:—

Bore	8 in.
Stroke	15 in.
Maximum B.H.P.	8.
Fuel consumption (pounds per B.H.P. hour)	0·919 lb.
Brake thermal efficiency	14·4 per cent.

In 1898 Professor Robinson tested a 25 horse-power Hornsby oil-engine with the following results:—

Bore	14·5 in.
Stroke	17 in.
R.P.M.	203.
Piston speed	576 ft. per minute.
Maximum B.H.P.	26·74.
Brake mean pressure (ηp)	37·2 lb. per square inch.
Compression ratio	3·4 : 1.
Mechanical efficiency	84·5.
Mean pressure	44 lb. per square inch.
Fuel consumption (pounds per B.H.P. hour)	0·72 lb.
Brake thermal efficiency	17·9 per cent.
Indicated thermal efficiency	21·2 per cent.

This latter appears to be the best result of any published test, but it is clearly very out of date, and presumably Messrs. Hornsby have been able to improve the efficiency of their engines since that time. The mean effective pressure in both cases is very low, and in another test carried out by Professor Capper on an 8 horse-power Hornsby engine, a mean pressure of only 28·9 lb. per square inch was obtained at full load, the brake thermal efficiency being only 14 per cent, and the indicated efficiency only 17·3 per cent. It would appear at first sight that engines of the Hornsby type should yield a fairly high mean pressure because:

1. The incoming air is not heated.
2. It is not necessary to restrict the area of the inlet port in order to obtain a high velocity for the atomization of the oil.

It seems probable that the mean pressure is limited by the fact that in such engines only the air actually contained in the hot bulb can be carbonized and burned; and that the remaining air in the cylinder remains inert. If this were the case, however, a very high relative efficiency might reasonably be expected. In Professor Robinson's tests the indicated efficiency was only 21·9 per cent, with an air-standard efficiency of 39·5 per cent, corresponding to a relative efficiency of only 55·5 per cent.

It is much to be regretted that so little information is available as to the performances of Hornsby oil-engines, because the system is an extremely interesting one. It is particularly interesting in view of the recent success of the semi-Diesel type of engine, which is but a very slight modification of the original Hornsby-Ackroyd

engine. In fact, the possibility of operating upon the semi-Diesel cycle is actually suggested and outlined in one of the patent specifications covering this engine, but does not appear to have been followed up.

The National Oil-engine.—The National oil-engine, built by the National Gas Company of Ashton-under-Lyne, near Manchester, is a comparatively recent design, and is evidently inspired by the Hornsby-Ackroyd principle. The general design of the engine is shown in fig. 177, from which it will be seen that the usual standard gas-engine practice is adhered to. The cylinder is partly overhung and partly embedded in the main frame, a feature which is common to all National engines of the horizontal type. The peculiar feature of this engine lies in the arrangement of the vaporizer and method of ignition, shown in fig. 178. The combustion chamber is made in the form of a long horizontal tube, the portion near the cylinder being parallel and water-jacketed, and the farther portion conical and unjacketed. At the extreme end of this chamber a small igniter is fitted, consisting of an internal open-ended tube. This tube extends nearly across the narrow end of the vaporizer. Immediately below the open end of the ignition tube is a cored passage leading back to the cylinder, as shown in the section. The piston is provided with a projecting piece, of smaller diameter than the cylinder, which fits into a counterbore formed in the cylindrical part of the combustion chamber. The action appears to be as follows.

During the suction stroke, pure air is taken into the cylinder through the main inlet valve, which is placed near the open end of the combustion chamber. At the same time, oil and air are drawn into the unjacketed portion of the chamber. During the early part of the compression stroke the oil in the unjacketed portion is vaporized, but has not sufficient air for combustion.

As the compression proceeds, air is driven along the parallel portion of the combustion chamber towards the oil vapour situated at the end; but, owing to the formation of this chamber, the gases are stratified, and very little diffusion takes place between the oil-vapour and air. Just before the completion of the compression stroke the projecting portion of the piston enters the open mouth of the combustion chamber, with the result that the air contained in the annular space is entrapped, and forced at a high velocity through the connecting passage into the farther end of the combustion chamber. Here it produces violent turbulence, and simul-

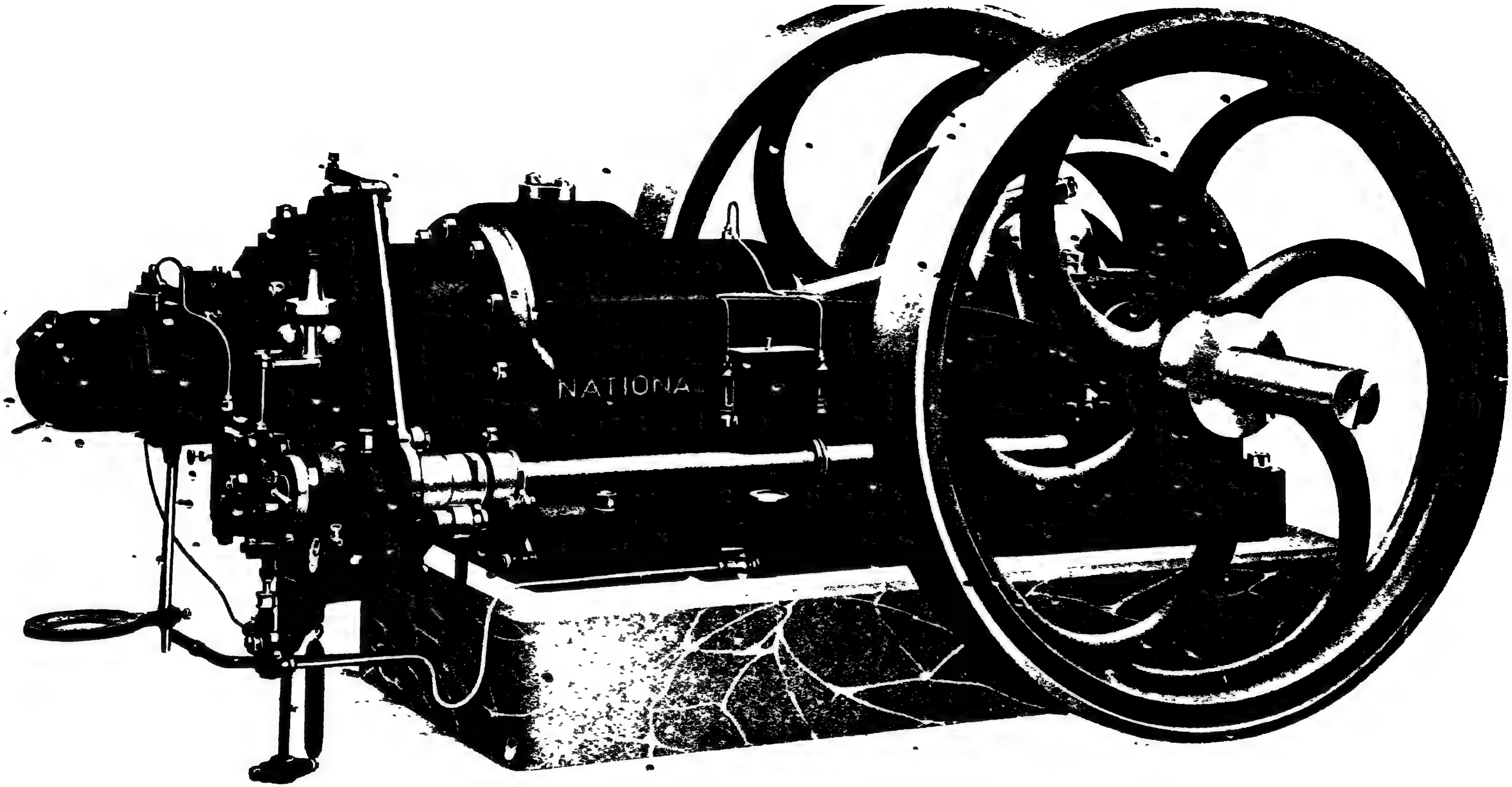

Fig 177 —35- to 50-B.H.P. National Oil-engine

taneously mixes the oil-vapour and air, bringing them into contact with the ignition tube, from which they are ignited. The whole arrangement appears to be an excellent one, for, although the gases are stratified, and therefore more or less stagnant during the compression stroke, they are thrown into a state of violent turbulence prior to ignition.

This arrangement would appear to have an advantage over the Hornsby system in that the temperature of the bulb itself is of less importance, since it controls vaporization only and is not concerned with the time of ignition as well, the latter being determined by the point at which the main entry to the combustion chamber is cut off by the projecting portion of the piston. With this system there appears to be no reason why a comparatively high compression ratio

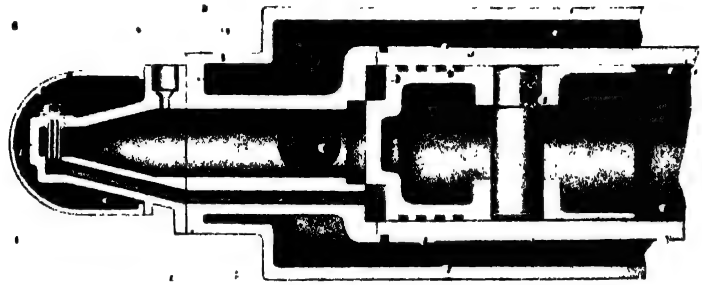

Fig. 178.—National Oil-engine: Combustion Head

should not be adopted, for if the working fluid be sufficiently stratified, there need be no mixture of combustible proportions in contact with the heated portion of the combustion chamber, and the main air supply escapes all pre-heating either within or without the cylinder. The whole arrangement appears to offer a great many advantages, and is decidedly attractive.

The author has been unable to obtain any information as to the extent to which the small passage in the ignition-tube opening becomes carbonized up, but there seems to be no reason why this should take place to any excessive degree. The engine is governed on the "hit and miss" principle, the governor acting upon the small pump which delivers oil to the vaporizer. No provision is made for maintaining a uniform temperature in the vaporizer, and it is said that this is unnecessary even when running on the lightest loads. It is obvious that if the ignition be timed, as in this case, by the motion of the piston, it must in any case take place at some period before the end of the compression stroke. Although the rate of inflammation may be comparatively slow, there is always

a certain risk of the pressure rising too rapidly, especially on heavy loads, when the temperature of the vaporizer is high. To obviate this, water injection is employed on the heavier loads, the function of the water being to retard the rate of inflammation, and, at the same time, to cool the bulb.

A test which the author witnessed on a small National oil-engine, of $5\frac{1}{2}$ horse-power, gave a fuel consumption of 0·69 lb. per B.H.P. hour, corresponding to a brake thermal efficiency of 19·2 per cent. Whether this is an exceptionally good result for this particular type of engine the author is unable to say, but the design is clearly one from which a comparatively high thermal efficiency may be expected.

The New Blackstone Engine.—A very interesting engine has recently been placed upon the market by the Blackstone Company, and is known as a "cold starting" oil-engine. In this engine, pulverization alone is relied upon, and the fuel is driven into the cylinder by means of a high-pressure air-blast. By this means the fuel can be so finely pulverized as to be readily combustible and all preheating is avoided, consequently a fairly high compression can be safely used, and also a high mean pressure. These advantages, however, are obtained at the expense of a good deal of extra complication in the form of a high-pressure air-compressor, always a costly piece of apparatus, and none too reliable at the best of times. The engine, however, is extremely interesting, and time alone can show whether it will be able to hold its own in competition with the other forms of liquid-fuel internal-combustion engines.

The engines described above represent only a very small proportion of those on the market at the present time, but they may be taken as fairly representative. In view of the fact that this class of engine is being rapidly superseded by the much more efficient semi-Diesel engine, it is not proposed to devote more space to it. In the somewhat specialized field of motor-boating, a very large number of vaporizing oil-engines are, however, still used.

CHAPTER XXVII

THE DIESEL ENGINE

The Diesel oil-engine has, during the last few years, assumed a position of very great prominence in the public eye, and has largely eclipsed many of the other types of internal-combustion engines, especially in the lay mind. This is largely due to its undoubted merits, but it is also to be accounted for by an extraordinarily well-organized and comprehensive campaign of advertisement throughout the whole of Europe, not only in the technical, but also in the lay press, with the result that the Diesel engine has assumed a degree of importance in the public mind to which it is not entitled. In the following pages, therefore, the writer will endeavour to put the merits and demerits of the Diesel as impartially as possible, without magnifying the former or ignoring the latter. By dint of careful advertising the public has been lured into the belief that the Diesel engine will ultimately replace the steam and the gas-engine as a prime mover, not only for stationary purposes, but also for marine and locomotive work. There is a general belief that the efficiency of the Diesel engine is something like double that of the gas-engine, that it can be constructed in any size, and up to any power, and that there is no purpose for which it is not suited.

In "booming" this type of engine its partisans have made a clever use of two arguments:—(1) That since the fuel is not admitted to the engine cylinder until the end of the compression-stroke, therefore pre-ignition, the bugbear of the explosion type of engine, cannot occur. (2) That the indicated thermal efficiency is from 45 per cent to 50 per cent. and, therefore, enormously greater than that of any other known form of prime mover.

Neither of these arguments is really convincing to an engineer who has seriously studied the subject. In the first case, pre-ignition can, and does, occur if by any chance the fuel finds its way into the cylinder, either through a missfire or a leaky or sticky fuel-valve;

and when it occurs, it is much more dangerous than in an explosion engine on account of the very much higher compression-pressure which is employed. Further, the indicated thermal efficiency of an engine is of purely scientific interest. It is the net, or brake, thermal efficiency which alone is of any consequence. Now the net thermal efficiency of the Diesel engine is but little higher than that of the gas-engine, the difference being only about 10 per cent in favour of the Diesel. Moreover, the indicated thermal efficiency of the Diesel engine is, even from the scientific point of view, of merely fictitious value, because air which has been compressed externally, and not recorded by the indicator, is admitted to the engine cylinder during the expansion stroke, where it appears on the credit, but not on the debit, side of the account. The high apparent indicated thermal efficiency argument has been used to its utmost, especially with shipbuilders, who are accustomed to reckon the power of their steam-engines on the indicated, and not on the shaft horse-power, with the result that comparisons are drawn which are grossly unjust to the steam-engine.

Again, it is generally believed that the Diesel engine works with far lower maximum temperatures than the gas-engine, and that, as a result, it can be built in much larger powers without the danger of structural failure due to the stresses set up by the temperature-gradient through the cylinder walls. This belief is entirely unfounded. It is true that, for equal mean pressures, the maximum temperature of the gases in a Diesel engine is slightly lower than in a gas-engine. On the other hand, owing to the very much higher pressures that might be set up in the event of pre-ignition, it is necessary to employ much thicker cylinder walls for the same size of cylinder than in an explosion engine. Hence the stresses due to temperature-gradient are greater, with the result that the limit of cylinder diameter is reached earlier in the Diesel than in the gas-engine. Gas-engines, with cylinders up to 51 in. diameter, are in daily use, and have been a commercial possibility for several years past; but, up to the beginning of 1915, no Diesel engine has been put into operation with cylinders larger than 32 in. diameter, though a few experimental engines have been constructed with slightly larger cylinders.

In spite of this fact there have been, from time to time, outcries in both the technical and lay press against the Admiralty for not experimenting with Diesel engines for the propulsion of the latest types of battleships and battle-cruisers. Such outcries show how

little the limitations of the present-day Diesel engine are understood.

The high thermal efficiency of the Diesel engine is recognized on all sides; but its comparatively low commercial efficiency in a country such as this, which is not oil-producing, is not always appreciated. The fuels suitable, and generally used, for Diesel engines are the better quality residual oils, i.e. crude petroleum after the petrol and paraffin have been removed; and, to a limited extent, tar oils. Now, neither of these are any longer waste products; on the contrary, both have a high and steadily rising market value. In 1914 the cost of residual oils in this country was in the neighbourhood of 60*s*. per ton. Taking the average fuel consumption of a Diesel engine as 0·45 lb. per B.H.P. hour, the cost of fuel per B.H.P. hour becomes 0·144*d*. With a steam plant, using coal at 16*s*. per ton, and consuming 1·5 lb. per B.H.P. hour, which is about the consumption of a medium-sized steam turbine of, say, 2000 B.H.P., the cost of fuel becomes 0·129*d*. per B.H.P. hour. Again, with a gas-engine and producer using coal at 16*s*. per ton, and consuming 1 lb. per B.H.P. hour, the cost of fuel per B.H.P. hour becomes 0·086*d*.

The cost of fuel in the three cases is, therefore:—

	Pence, per B.H.P. Hour.
Gas-engine and producer (bituminous) .	0·086
Steam turbine	0·129
Diesel engine	0·144

These figures, however, are based on the assumption that the engines are working at full power in all cases, a condition which places the Diesel engine in the most unfavourable light, because it takes no account of two very valuable properties which the engine possesses: (1) The fuel consumption per B.H.P. hour is practically the same between half-load and full-load; (2) it can be started instantly from cold, hence there are no stand-by losses.

In order to view the matter in a fair light, it is necessary to compare the cost of fuel when all three types are working under commercial conditions.

Diesel Engines for Power-station Work.—Let us consider the conditions of operation of a plant required to deal with a peak load of, say, 6000 B.H.P., and with a load-factor of 30 per cent, which is a not uncommon condition in a small electric-power plant. The efficiency of both gas and Diesel engines is practically

unaffected by the size of the individual units; therefore, from the point of view of fuel consumption alone, it will pay to use comparatively small units, say 10 units each of 600 horse-power, not counting reserves. In the case of the steam plant the efficiency depends very much upon the size of the unit, and therefore it will be desirable to use the largest units consistent with the load factor, say 3 units each of 2000 B.H.P.

Now these conditions all favour the Diesel engine, because there will be no stand-by losses, and owing to the large number of units no single unit need be run at an inefficient load. Therefore the cost of fuel may again be taken as 0·144*d.* per B.H.P. hour.

The same applies, but to a much lesser extent, to the gas-engine, for there will be stand-by losses in the producers, which must be kept banked in readiness to meet the peak load. The coal consumption in this case may be taken as 1·3 lb. per B.H.P. hour, or 0·112*d.* per B.H.P. hour.

In the steam plant the stand-by losses will be considerable, owing to the large size of the units, which cannot always be run at, or near, their most efficient load. Sufficient boilers must always be banked in readiness to meet the peak load, and one unit must be kept running light, or at least warmed up for considerable periods, because steam turbines cannot be started quickly from cold. The fuel consumption under these conditions will probably average about 2·5 lb. of coal per B.H.P. hour, or 0·215*d.*

The cost of fuel under commercial conditions will therefore be approximately:—

	Pence per B.H.P. Hour
Gas-engine and producer plant .	0·122
Diesel engine	0·144
Steam turbines	0·215

These figures are based upon the cost of fuel alone, and the gas-engine now appears in the most favourable light. If an ammonia-recovery plant were provided the cost per B.H.P. hour of the gas-engine plant would be still further reduced, but the cost of upkeep of gas-engines and producers is considerable. That of Diesel engines is less, and of steam-turbines less still. From the point of view of *capital* outlay the Diesel plant is probably the most expensive, with the gas-engine second, and the steam-engine third; but when ground rent, buildings, &c., are taken into account there is not very much to choose between all three,

and much must necessarily depend upon local conditions, water-supply, &c.

It is not proposed to deal further with the question of comparative running costs, but merely to indicate that, from the important point of view of commercial efficiency, the Diesel engine is inferior to the gas-engine *with the present price of fuel oil in this country*. In Russia, and in other countries where oil is abundant and coal expensive, these figures would be very different. The above figures are not intended to discredit the Diesel engine in any manner, for this engine can justly claim the distinction of being the most efficient prime-mover ever produced, but merely to show that its commercial possibilities are not unlimited.

Its principal *assets* are—

1. It is compact and self-contained.
2. It can be started instantly from cold.
3. Its high efficiency is maintained over a wide range of load.
4. It can use residual oils or tar oils without the necessity for a gas-producing plant.

These are all substantial advantages, which render the engine admirably suitable for a great variety of purposes.

Its principal *disadvantages* are:—

1. Its high first cost and complication.
2. That its use, as compared with the steam-engine, is limited to comparatively small powers.
3. The choice of fuel is restricted to petroleum and coal oils, both of which are no longer waste products, and whose market value is liable to violent fluctuations. It is true that the Diesel engine will also run on animal or vegetable oils, but at the present time such oils are more expensive than mineral.

The general principles of the Diesel cycle and its efficiency have been dealt with previously.

Credit for the development of this engine is due both to the late Dr. Rudolf Diesel and to the Augsburg branch of the Maschinen-Fabrik Augsburg-Nurnburg, who showed very great insight and perseverance during the early experimental stages. The mechanical difficulties to be contended with were very serious, and it was several years before anything approaching a commercial design could be evolved. As originally conceived by Dr. Diesel it was

intended to employ isothermal compression during the first part of the compression stroke, and adiabatic compression during the second part. The expansion stroke was to be isothermal during the first part, the necessary heat required being obtained by the admission of coal dust to the cylinder, and the temperature kept constant until a certain point in the stroke was reached. Then the supply of fuel was to be cut off, and adiabatic expansion continued until the end of the stroke.

This cycle, which is practically the Carnot cycle, of course, provides a very high efficiency, but it is hardly a practical one, on account of the extremely low mean pressure obtained. Early experiments soon showed that the use of coal dust as fuel was impossible, and the constant-temperature cycle was soon changed for one employing constant pressure, which gave a far higher mean with the same maximum pressure; but, of course, a somewhat lower efficiency. It was also found that in order thoroughly to pulverize and distribute the fuel in the very short space of time available, it was necessary to inject it with compressed air. This necessity for the use of the high-pressure compressed air has always been, and still is, a weak point, for not only does it add to the cost and complication of the engine, but the compressor is, in itself, a source of weakness and unreliability.

Fuel Injection.—In the Diesel engine, as at present constructed, whether operating on the two- or four-stroke cycle, air alone is drawn into the cylinder, and is compressed adiabatically to a pressure of from 450 to 500 lb. per square inch. This compression raises the temperature of the air sufficiently to ignite with certainty practically any fuel which the engine is intended to use. At the inner dead-centre, or, in practice, a few degrees before it, a small needle-valve is opened, and a charge of fuel is blown into the cylinder by means of the highly-compressed blast air. Owing to the extremely high pressure of this air, generally from 700 to 1000 lb. per square inch, the fuel is thoroughly pulverized, and enters the cylinder as a fine mist, which ignites as soon as it comes into contact with the highly-heated air in the combustion-space. The rate of admission of the fuel, and consequently the temperature, is so adjusted that the pressure is maintained constant during the first period of the outward stroke of the piston. The fuel-valve is then closed, and the highly-heated air and products of combustion are expanded until the end of the stroke.

It is evident that to obtain the best efficiency, it is of the

utmost importance that the fuel shall be so finely pulverized and distributed, that each particle shall come in contact with the necessary quantity of oxygen, and shall be burnt before it can reach the cold walls of the cylinder, and before the expansion commences. Apart from the loss of efficiency, any unburnt oil that comes in contact with, and deposits upon, the cylinder walls, will either burn slowly on the surface, throwing down a deposit of carbon, or will mix in a partially burnt condition with the lubricating oil on the cylinder walls, and so interfere with the piston lubrication, and cause gumming of the piston-rings.

Effect of Valve Leakages.—The fuel-valve is generally supplied with fuel oil by means of a small plunger pump, which delivers the oil to the valve some time before it is required. It is allowed to remain there until the valve is opened, when it is driven into the cylinder by means of the highly compressed air. The valve itself is always under air pressure, and it is, therefore, very important that it shall be kept tight, for any leakage will involve not only the loss of blast air, but, also, some of the fuel oil will be driven into the cylinder as soon as it reaches the valve, and at a time when it will only increase the negative work. If the valve should stick open from any cause, the whole of the oil will be driven into the cylinder as soon as it is delivered from the pump, and it is conceivable, though highly improbable, that it *might* be driven in just before the end of the compression stroke, and so result in a very serious pre-ignition.

Again, if either the fuel or air-starting valves should, from any cause, remain open so as to admit air to the cylinder during the compression stroke, it is clear that supercharging of the cylinder will take place, and the compression pressure may be raised to a dangerous figure. The rise of pressure due to this source is, however, comparatively gradual, and can be guarded against by the employment of a relief valve adjusted to blow off at a pressure of, say, 600 lb. per square inch. Such a valve, however, owing to its inertia, will not be able to relieve very sudden rises of pressure, such as occur in the event of pre-ignition, to any great extent. Finally, it is clear that, under certain combinations of circumstances, it is possible for the fuel valve to remain open, and thus produce both supercharging and pre-ignition, and this is the greatest danger of all.

Leakage must, at all costs, be avoided in a Diesel engine, because this engine depends solely upon the adiabatic compression of the

air to provide the necessary heat to ignite the fuel. Should there be any serious amount of leakage during the compression stroke, the temperature of the air may not be sufficient to ignite the fuel. Under these circumstances the fuel will be partially vaporized, and, unless all the vapour be expelled during the following exhaust stroke, it may be ignited during compression, thus causing a serious and often dangerous pre-ignition. To avoid this risk of leakage very great care must be taken in the manufacture of such parts as pistons, liners, and valves, for not only must they be machined with extreme accuracy, but they must also be thoroughly annealed in order to avoid deformation when heated. All this, of course, adds to the cost of the engine.

Compression Ratio.—It has already been pointed out that the air-standard efficiency for a constant-pressure or Diesel engine is dependent, not upon the ratio of compression alone, but also upon the maximum temperature. The air-cycle efficiency diminishes as the temperature (and therefore the mean pressure) increases. The ideal efficiency for the actual working fluid diminishes with increase of temperature to a still greater degree on account of the increase in the specific heat of the gases, and the actual efficiency obtained from test figures shows the same decline. It has also been pointed out that for equal compression ratios the air-standard efficiency of the Diesel engine is lower than that of an explosion engine under all working conditions, that the air-standard efficiency of the latter is independent of the temperature, and that the efficiency of both is the same at the point of no-heat supply.

In an explosion engine the ratio of compression employed is the highest that will ensure against spontaneous ignition of the gases due to the heat of compression, even when the engine is hot and dirty. In the Diesel engine the ratio generally employed is the lowest that will ensure spontaneous ignition, even when the engine is clean and cold. In both cases the actual figure employed is dependent to some extent upon the nature and ignition-point of the fuel. From the point of view of indicated thermal efficiency alone, it is obvious that the higher the compression the higher the efficiency; but the increase in efficiency between, say, a 13:1 and a 15:1 compression ratio is comparatively small, being only about 2 per cent. The increase in maximum pressure between the two ratios in the event of pre-ignition is about 30 per cent. Now it is clear that the mechanical efficiency, to say nothing of the cost, is dependent upon the ratio of the maximum abnormal pressure to the

mean pressure, because the mechanical efficiency depends mainly upon the weight of the reciprocating parts, which, in turn, are governed by the maximum pressures. Between a ratio of 13:1 and 15:1 the difference in mean pressure is inconsiderable, but the difference in maximum pressure is very large. Consequently, the higher compression pressure would bring about such a reduction in the mechanical efficiency that the net thermal efficiency would probably be actually reduced, while the weight and cost of the engine would be increased almost in proportion to the increase in maximum pressure. From the above consideration it is clear that it is not desirable to increase the compression pressure in a Diesel engine above the minimum which will ensure ignition of the fuel.

CHAPTER XXVIII

FUEL INJECTION

It is obvious that the practical success of a Diesel engine must depend very largely upon the pulverization and distribution of the fuel, and attention must be concentrated upon the fuel-valve, whose function it is to control: (1) the rate of admission; (2) the degree of pulverization; and (3) the distribution of the fuel within the combustion-space.

1. **Rate of Admission.**—In the ideal constant-pressure cycle the fuel is admitted at such a rate that its combustion maintains a constant pressure within the cylinder, throughout the full period of admission. In practice, no fuel-valve will do this, nor is it desirable that it should do so. If the admission of fuel be too rapid, the pressure within the cylinder will rise above the compression pressure, and, within reasonable limits, this is a desirable feature, for the cycle then approaches more nearly to the constant volume cycle, and the efficiency is, in consequence, improved. The pressure of compression is, however, already so high that any further considerable rise may prove destructive to the bearings. It is a comparatively easy matter so to design the fuel valve that the pressure shall be maintained within safe, on the one hand, and, on the other hand, economical limits, provided that the viscosity of the oil fuel remains the same. Unfortunately, the viscosity varies considerably, not only with different samples of fuel, but also with the same fuel under varying temperatures. In an engine such as a stationary engine, which runs at a constant speed, it is not difficult so to adjust the fuel valve that the pressure shall remain approximately uniform when the engine has reached its normal temperature, and so long as the brand of fuel used remains the same. In an engine required to run at variable speeds, however, the difficulty is very great indeed, because if the rate of fuel admission remains constant on a time basis, it is obvious that a given quantity will be admitted earlier in the stroke, at low speeds, and later, at high

speeds. Thus in the one case the pressure may rise to a dangerous degree, and in the other, combustion may be so far delayed as to cause a considerable loss of efficiency. Again, at low speeds, the time during which the fuel is lying in the hot valve chamber is longer, hence it takes up more heat, and its viscosity is reduced. This will result in still more rapid injection, and a still further rise of pressure.

All these conditions can be compensated for to some extent by varying the blast-air pressure, which is largely instrumental in controlling the rate of admission, but this requires very careful adjustment. In marine engines, a reduction in speed is always accompanied by a corresponding reduction in load, and, since the ratio of mean pressure to speed is definite, it is easy to vary the air pressure in accordance with the engine speed. Up to the present time, Diesel engines have not been used for motor traction, nor does there seem any prospect of their being so employed, for the sudden changes, both of speed and load, that are demanded of a traction motor could not be dealt with by a Diesel engine, unless some entirely new method could be devised for regulating the rate of admission of the fuel.

2. **Pulverization of Fuel.**—The degree of pulverization is, of course, of primary importance. It is obviously essential that the whole of the fuel shall be divided into such extremely small particles that each can vaporize and burn completely during the very small space of time available. The fuel issues from the valve in the form of extremely fine globules, travelling initially at a very high velocity, corresponding to the pressure difference between the blast-air and the cylinder pressure. Owing, however, to the small weight of the particles, their inertia is small, and their initial velocity falls very rapidly. On coming into contact with the highly-heated air within the cylinder, these globules begin to vaporize on the surface. The vapour thus formed ignites on coming into contact with the oxygen of the air, and from then on the particles burn until they are completely consumed.

Each globule of oil must be considered as a sphere of liquid immediately surrounded by an atmosphere of pure oil vapour. At a greater radius, the atmosphere consists of burning oil vapour and air. The liquid in the centre of the sphere is rapidly vaporizing, due to the heat radiated by the outer layer of burning vapour, and this continues until the whole of the central core of liquid has been vaporized and burnt. The rate of burning depends upon the diameter of each globule, provided, of course, that it is surrounded

by sufficient air to maintain combustion. If the diameter of the globule be so great that it can reach the comparatively cold walls of the cylinder before it is completely burnt, it will simply adhere to these walls and burn slowly from the surface. Owing to the cooling action of the cylinder walls, the heavier fractions of the fuel will not be vaporized at all, and there will remain a deposit of unburnt carbon which will gradually accumulate. If the surface be above a certain temperature, much the same thing will occur: in this case, however, the carbon will not adhere to it, but will flake off.

From the above considerations, it is clear that it is of the utmost importance (1) to pulverize the fuel as finely as possible, and (2) so to direct the spray that it shall not come into contact with any cold surface before it is completely burnt. On account of the very high compression ratios employed in Diesel engines, the thickness of the layer of air in the combustion space is necessarily very limited, and the distance which particles of fuel can travel without coming in contact with some metallic surface is very small. For this reason, the Diesel engine, more than any other type, depends for its efficiency upon the employment of a relatively long stroke.

3. **Distribution of Fuel.**—The distribution of the fuel throughout the combustion space is of even more importance than the pulverization. However finely the fuel may be pulverized, complete combustion is impossible unless each particle of fuel is surrounded by the necessary quantity of air. Many fuel valves which give excellent pulverization are found, in practice, to give poor results, because the individual particles of oil are not sufficiently separated from one another to allow room for the necessary air for combustion. At the same time a considerable proportion of the air in the combustion space never comes into contact with the particles of oil, and is completely inert. Under these conditions the engine will smoke heavily when fully loaded, due to the incomplete combustion of the particles of oil. The ideal fuel valve is one which will direct a spray of oil into every corner of the combustion space, so that the whole of the air is thoroughly impregnated, and at the same time the spray is nowhere so dense that it cannot find sufficient air for complete combustion.

It is, of course, needless to say that this condition is never realized in practice. If it were, the mean effective pressure would be in the neighbourhood of 200 lb. per square inch. It is probable that the nearest approximation to the ideal is reached in the experimental Junkers engine, to be described later. In this engine the

ratio of diameter to length of the combustion space is approximately as 3·2 : 1, and two fuel valves are employed directing the sprays tangentially, in order to produce a whirling motion. It is obvious that the distribution of the particles of fuel is greatly helped by any turbulence within the combustion space, and this turbulence is largely produced by the high-pressure blast-air which enters with the fuel.

It is owing to considerations of the distribution of the fuel which have led designers of Diesel engines to adhere strictly to the simplest and most symmetrical form of combustion space, namely, that in which the cylinder cover is merely a flat plate, with the fuel valve mounted in the centre, and the inlet and exhaust valves on either side. In order to leave a sufficiently deep layer of air immediately below the fuel valve, the top of the piston is generally made concave. The use of side pockets for the valves is inadmissible, unless the fuel valve can be so placed that the spray will reach these pockets; otherwise, they will simply contain dead air, and the power of the engine will be reduced in consequence. The flat combustion head is by no means the only possible form, and, in horizontal engines, the conventional gas-engine design is perfectly practicable, the only dead air being that entrapped between the piston and the open end of the combustion chamber, which can easily be reduced to a very small proportion. It can readily be seen that the difficulties in the way of producing a satisfactory double-acting engine are enormous, for the presence of the piston-rod reduces the shape of the combustion chamber to that of a ring. Reasonably good distribution can, however, be obtained by the use of two valves, admitting the spray tangentially, or by so designing the cylinder that, at the end of the compression stroke, the whole of the air is compressed into two valve-chambers of sufficient capacity to accommodate it. In this case, of course, each must be fitted with its own fuel valve. The problem of the double-acting engine, however, is still further complicated by the necessity for using a stuffing gland, which is always liable to leak, and leakage of compression in a Diesel engine means not only loss of efficiency, but is in itself a very real danger.

The Air Blast.—In all Diesel engines as yet, with the exception of the Vickers submarine engines, the fuel is pulverized and distributed by means of compressed air. The advantages of this are:

1. A very much finer degree of pulverization can be obtained than is possible by mechanical means.
2. The particles can be more thoroughly distributed.

3. The particles are surrounded initially by a certain amount of air for combustion.

4. The turbulence produced by the inrush of the air accelerates combustion, and stirs up any dead air in the cylinder.

The disadvantages are:

1. It is necessary to compress the blast or injection air to a very high pressure, generally from 700 to 1100 lb. per square inch; and the compression of air to such pressures involves the absorption of a considerable amount of power only a portion of which is returned as work done in the cylinder.

2. The high-pressure compressor is in itself costly, complicated, a source of weakness, and even of danger.

3. The expansion of air through the fuel valves, which is not truly isothermal, involves a reduction of temperature which is detrimental to combustion; and, on light loads, if the proportion of air be large it may prevent ignition altogether.

The first two objections are the really serious ones, and the second has been brought into peculiar prominence lately by the failure of the blast-air compressors in Diesel-engined vessels, and several fatal accidents due to the bursting of air-pipes and receivers.

Compressor Efficiency.—The proportion of power absorbed by the blast-air compressor depends upon the nature of the fuel and the load, but is generally from 3 to 8 per cent of the indicated horse-power. The compression of air to a pressure of from 800 to 1100 lb. per square inch must inevitably be a very inefficient process. If a two-stage compressor be employed, the number of compressions in each stage ranges from 5 to 7, and if the compression within each stage is assumed to be adiabatic, and the intercooler between the two stages so efficient that the air is cooled down to its original temperature before passing into the high-pressure stage, then the efficiency of the compressor, as compared with the true isothermal compression, will be in the neighbourhood of 66 per cent. This shows a loss of 34 per cent due to the heat of compression. To this must be added the valve losses, which are quite considerable, and the mechanical friction of the compressor.

The overall efficiency of the blast-air compressors, as used on Diesel engines, is probably little more than 45 per cent. It is not at all easy to say what proportion of the total indicated power of the compressor is returned as useful work in the cylinder of the engine, but it is probably not more than 60 per cent of the indi-

cated, and 40 per cent of the net, power absorbed by it. In spite of its low efficiency, however, the loss is rather more than compensated for by the better and more complete combustion, as compared with any method of mechanical injection that has been tried up to the present, for, owing to better distribution of the fuel, there is less dead air, and a higher mean pressure can therefore be employed.

Compressor Explosions.—Owing to the high temperature generated by the adiabatic compression of the air in the different stages of the compressor, and to the obvious necessity for lubricating it, there is always a possible risk of the lubricating oil being ignited during compression. Under normal working conditions, with a restricted supply of oil of high flash-point, this danger is remote, but in the event of partially choked or leaking delivery valves it becomes serious. Ignition of the lubricating oil does sometimes occur, generally in the pipe leading from the high-pressure stage to the after-cooler. It has occasionally been known to spread to the storage bottles, which may contain a considerable accumulation of oil carried over by the air. When this occurs, the resulting explosion is likely to be of a disastrous nature, though such an accident can generally be avoided by periodically blowing out any accumulation of oil in the storage bottles.

The effect produced on the temperature by a leaking delivery valve is not perhaps quite obvious at first sight, but by allowing highly heated air to pass back through the valve, during the suction stroke the suction temperature may be raised to a considerable degree, for the high-pressure air expands through the delivery valve without doing work, and consequently without any considerable loss of heat. It is obvious that if the suction temperature be increased by only a comparatively small amount the effect upon the compression temperature will be considerable, and quite a small leak may easily produce a temperature rise of as much as 200° F. at the end of the compression stroke, which may be sufficient to cause ignition of the lubricating oil. The effect of clogging of the delivery valves is, of course, obvious, in that it increases the resistance, and therefore both the degree of compression and the temperature.

Fuel Valve Design, Closed Type.—In the actual design of the fuel valve there is a wide diversity of opinion, but, broadly speaking, the types of valve generally employed may be divided into two classes, the closed and the open. A typical example of the closed type is given in fig. 186 (p. 428). In this arrangement the fuel oil is delivered to an annular chamber surrounding the valve spindle,

which is also in communication with the high-pressure blast air. In order to prevent the whole of the oil from being driven into the cylinder ahead of the air, a number of baffle-plates are provided, drilled with small holes. The oil falls on to these baffles, is driven through them, and carried into the cylinder by the rush of blast air that takes place when the valve spindle is lifted. The primary function of the baffle-plates is to act as a brake in order to prevent the oil from entering too rapidly, and the number of plates and the size of the holes in each must be adjusted to suit the viscosity of the oil. The baffle-plates also probably serve a useful part in mixing the oil and air. Below the seating of the valve is a separate plate, generally termed the flame-plate, having a small orifice through which the oil and air pass at a very high velocity, and by means of which the fuel is pulverized and spread out throughout the combustion chamber.

This is probably the simplest form of valve, and is one which has been found generally satisfactory, but it is, of course, open to improvement. In the first place, when once the number of baffles and the size of the holes have been decided upon, the braking effect will be dependent upon the quantity of oil passing through. This can be, and generally is, adjusted by varying the air pressure, either automatically or by hand. In some cases, however, the desired effect is produced by admitting the fuel lower down in the valve-chamber, thus short-circuiting a certain number of the baffle-plates on full load, but admitting it higher up on the lighter loads. Another method, and one which is fairly simple, is so to arrange the baffle-plates that every alternate one can be rotated through a small angle, in order to bring the holes directly under one another, or to stagger them. Yet another method is to drill a small hole, generally through the valve spindle itself, leading from above the baffles to a point just above the valve seating. The diameter of the hole is such that a portion of the fuel will pass through it. As the quantity is increased the space between the baffle-plates is first filled up, and then any excess of oil will flow down through the small hole to near the valve seating. In this manner the quantity of oil which is driven through the baffles remains practically the same at all loads.

Open-type Valve. In the open type of fuel valve, the fuel is admitted between the valve seating and the flame-plate. No baffles are used, and the rate of fuel injection is controlled by the rate at which it is delivered to the passage between the seating and the flame-plate. In this case, the fuel pump is timed to deliver its

supply during the first portion of the expansion stroke, which is a decided advantage, because, in the event of the valve sticking open, there is no fuel present to be driven into the cylinder until the extreme end of the compression stroke. In this way one of the possible sources of pre-ignition is obviated. The open type of fuel valve is naturally better suited to horizontal engines, because the passage leading from the valve seat to the flame-plate must necessarily be horizontal, for, in order to prevent a blast of cold air from entering the cylinder ahead of the oil, a certain proportion of it must be delivered, and must lie in the passage before the fuel valve is opened. It is most undesirable to have any changes of direction in this passage: hence it follows that the open-type fuel valve is generally used in horizontal, while the closed type is used in vertical engines.

"Ignition Oil."—For the use of tar oils, which have a very high ignition temperature, either an excessively high compression must be employed, which is by no means desirable, or a small quantity of oil, of low ignition-point, must be admitted ahead of the tar oil, and this latter is the method usually adopted.

Two fuel pumps are fitted, one delivering the tar oil to the fuel valve above the baffle-plate, while the second and smaller pump delivers gas oil, or some other oil with a low ignition-point, to the valve-chamber below the baffle-plates, so that when the valve spindle is raised the lighter oil enters the cylinder first. In the horizontal Deutz engines, an open-type fuel valve is used for tar oils. In this case the ignition oil is admitted to a small passage alongside the main passage, but communicating with the latter immediately behind the flame-plate. During the compression stroke the ignition oil is vaporized, and the vapour driven back into the main passage. At the end of the stroke the valve spindle is lifted, and the contents of the passage driven into the combustion space. By this means the ignition-oil vapour enters the cylinder first, and ignites, followed by the tar-oil and air.

The fuel pumps employed for the supply of oil to the fuel-valve chamber are of the ordinary plunger type, and are generally operated from eccentrics, mounted either on the camshaft or the intermediate shaft. The quantity of oil delivered is controlled by the governor, which operates upon the pump suction valve by holding it open during a certain period of the delivery stroke. This system of control has answered admirably, and is now almost universally adopted for Diesel engines.

CHAPTER XXIX

FOUR-CYCLE ENGINES

The Mirrlees Slow-speed Type.—The engine illustrated in fig. 179 is made by Messrs. Mirrlees, Bickerton, & Day, of Stockport, and is one of the most successful engines of its type on the market. It is rated at 50 horse-power when running at a speed of 250 R.P.M. In general construction it conforms to what might be described as the accepted Diesel practice, that is to say, the standards and the main body of the cylinders are all cast in one piece, forming a very rigid **A** frame. The cylinder-head is a separate casting, and is perfectly flat on the inside. The fuel valve is mounted centrally in the cylinder-head, with the exhaust and inlet valves on either side of it. The camshaft is carried on brackets attached to the **A** frame, and on a level with the cylinder-head, and all the valves are operated directly from it by means of short rocker arms. All this has now become standard practice, both in this country and on the Continent. As a design it was originated by the Maschinen Fabrik, Augsburg, and has been followed by practically every maker of stationary four-cycle Diesel engines, though just recently the horizontal four-cycle Diesel has put in an appearance, and it is very possible that this design will make considerable headway for stationary work.

The mechanical features of this engine are very well illustrated in the sectional drawings, figs. 180 and 181. The bedplate is a fairly simple iron casting, and calls for no particular comment. The crankshaft is carried in large white-metal-lined bearings, and each is provided with two ring lubricators, dipping into large troughs beneath them. A large inspection door is provided over each bearing, so that the rings can readily be inspected. It will be noticed that the spiral gearing driving the camshaft is fitted in the middle of one of the main bearings, an arrangement that makes for great rigidity and silence. The high-pressure air compressor for the blast-air is mounted on the same bedplate, and driven from a small

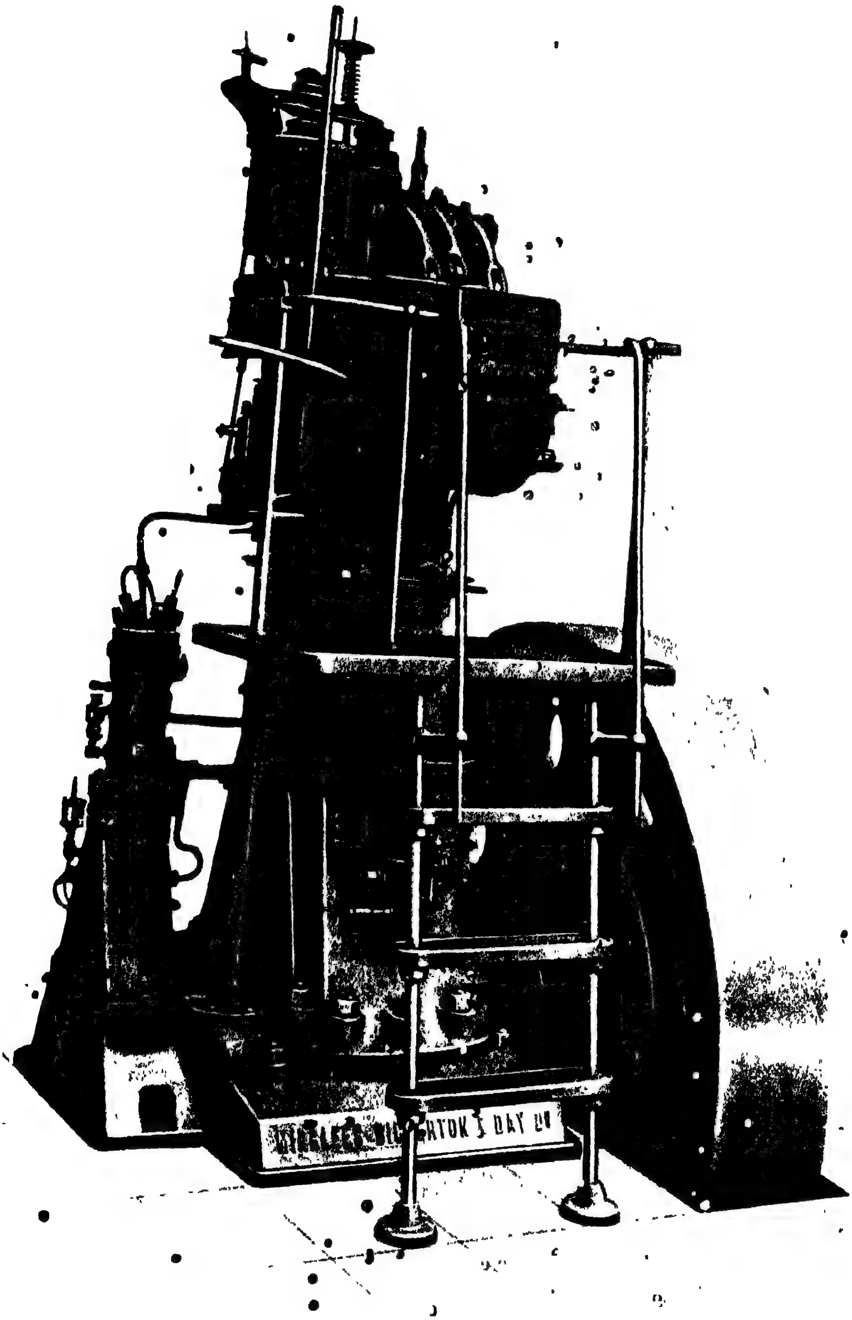

Fig. 179. 50 H P. Mirrlees Diesel Engine

overhung crank, shrunk or pressed on to the main shaft. The A frame has a very wide spread to ensure rigidity, and is amply ribbed below the cylinder liner. The latter is pressed in from the top, and

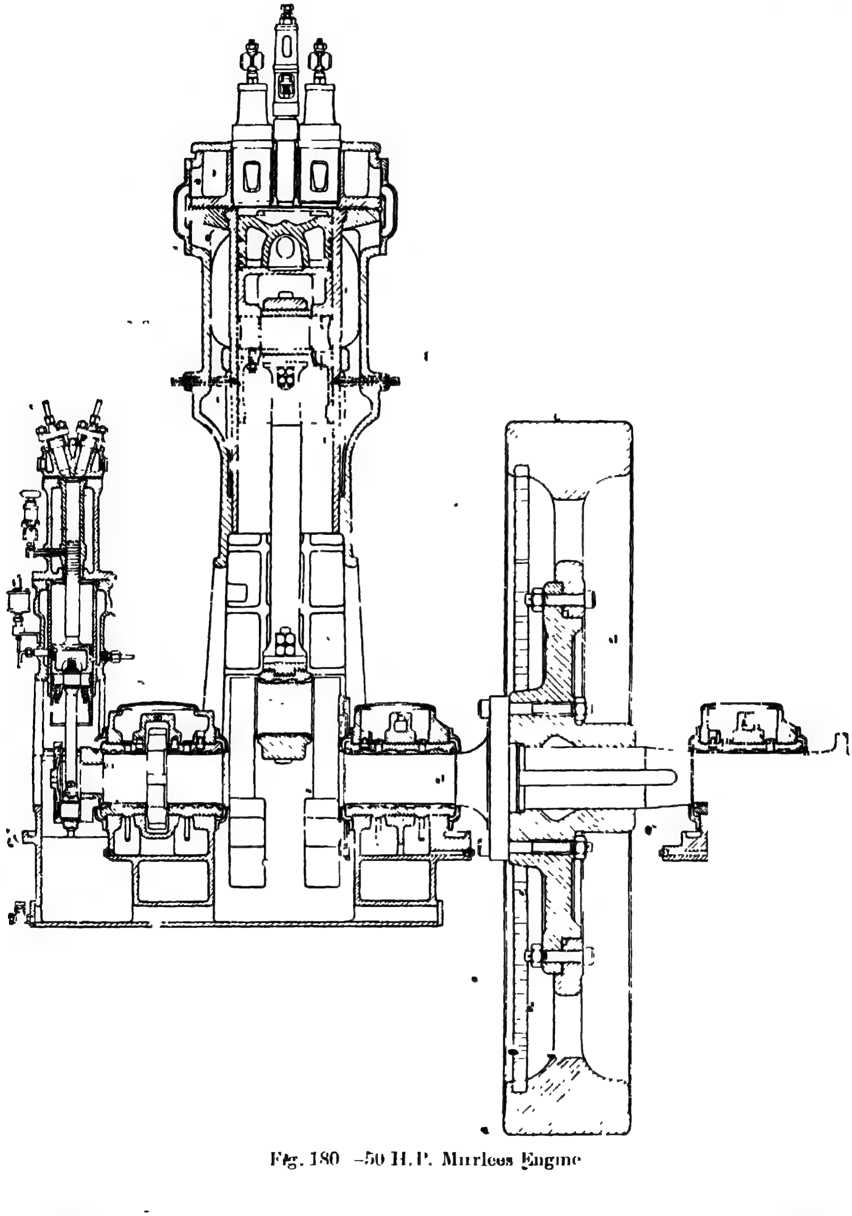

Fig. 180 —50 H.P. Mirrlees Engine

is guided both midway and at the bottom by means of internal flanges bored out to receive it. The lower end of the liner is made watertight by means of a rubber ring let into a groove, in accordance with the usual gas-engine practice.

The liner itself is a perfectly plain symmetrical casting. Great care is taken to avoid distortion, and after rough machining it is left to stand for some considerable time, in order to allow any deformation due to casting stresses to take place before it is finally

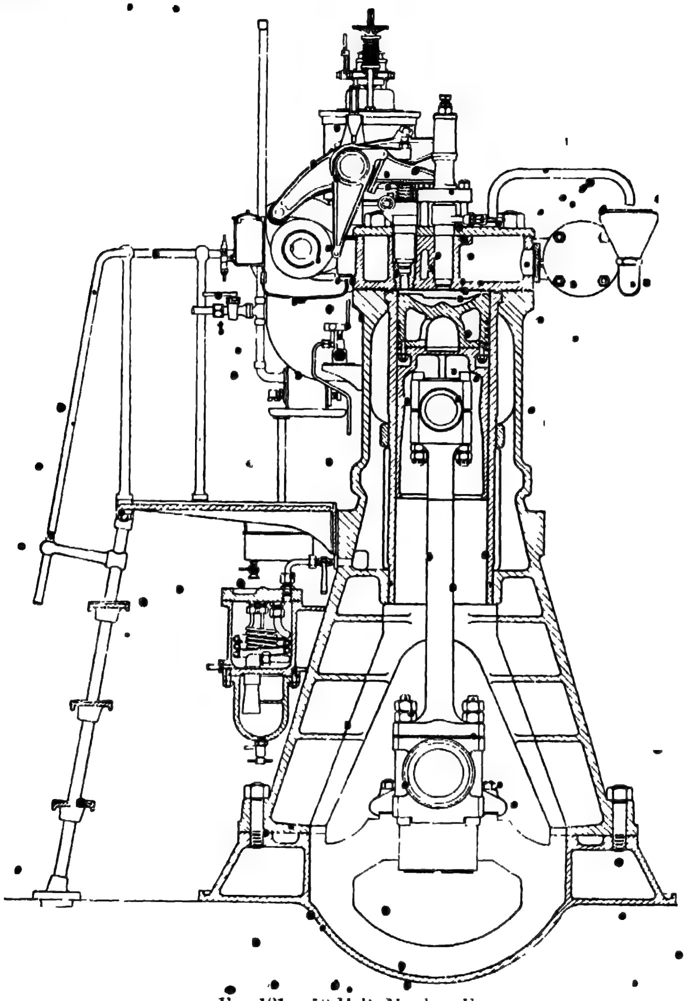

Fig 181.—50 H.P. Mirrlees Engine

machined and ground out dead to size. This is, of course, a matter of the utmost importance in an engine employing a compression pressure of about 460 lb. per square inch, and depending entirely upon the compression for the ignition of the fuel. In order to prevent any distortion of the liner due to the angular thrust of the connecting-rod, it is supported by the frame at a point slightly above the centre of travel of the gudgeon-pin, where the maximum

thrust occurs. The connecting-rod (fig. 182) is of the ordinary marine type, but the bearing "brasses" are stamped from mild steel.

The piston (fig. 183) is of cast iron, and is formed of two parts.

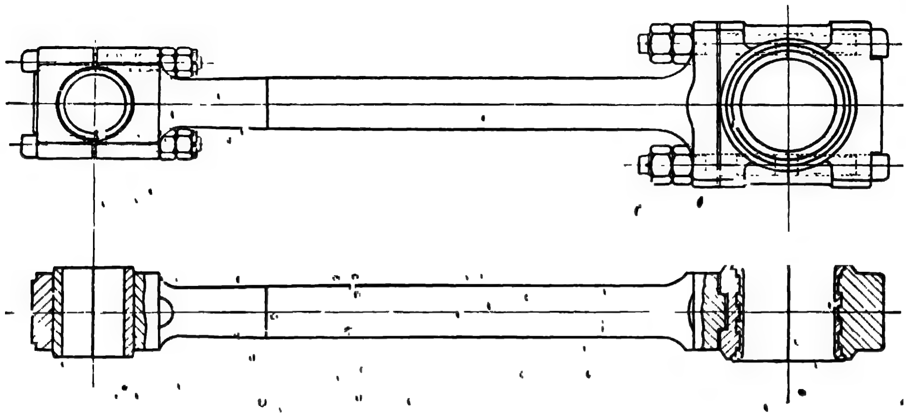

Fig. 182.—Mirrlees Engine Connecting-rod

The lower, or cross-head, portion is made a very close fit in the liner, and is slightly relieved at the sides to prevent binding, due to the distortion set up by the gudgeon-pin bosses. The upper portion, which carries the four piston-rings, is made quite a loose fit in the liner to permit of free expansion. The top piston-ring is fitted a considerable distance below the top of the piston, in order to protect it from the very high temperatures to which the piston is subjected, and so prevent it from becoming stuck fast in its groove with carbonized lubricating oil. The cylinder-head (fig. 184) is a plain iron casting, water-jacketed, and bored out to receive the four valves—exhaust, inlet, fuel, and starting. Each valve is provided with a separate detachable cage, so that it can readily be withdrawn for inspection or cleaning. It will be noticed that the exhaust, fuel, and inlet valves are fitted in a row across the top of the cylinder-head. This construction appears to be unavoidable, but it is decidedly objectionable for two reasons:

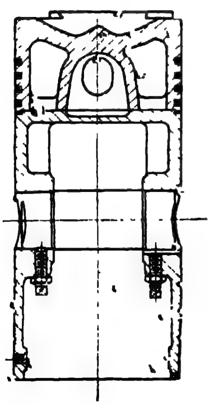

Fig. 183.—Mirrlees Engine Piston

1. It is not possible to water-jacket the head between the valves; consequently there is a wide belt of unjacketed metal just at the point of maximum heat flow, which must set up severe

stresses in the casting, and is liable, in larger engines, to cause fracture.

2. Neither the inlet nor the exhaust valve can be made of adequate size to ensure the best volumetric efficiency.

It is not at all easy to see how either of these objectionable features can be avoided without a radical alteration of the design; and it must be remembered that in a Diesel engine it is of special importance to keep the combustion chamber as compact as possible, in order that the fuel, as it issues from the fuel valve, shall be distributed evenly throughout the whole. In this connection the shape of the top of the piston should be noted. It is made cup-shaped, with a slight projection in the centre against which the fuel impinges and is spread out horizontally. The surface temperature of this projecting part is so high that the fuel will not readily adhere to it, as it would to the cool walls of the cylinder if it impinged upon them. In order to facilitate the removal of the valves the rocker arms are made in two pieces, bolted together in such a manner that they can easily be disconnected.

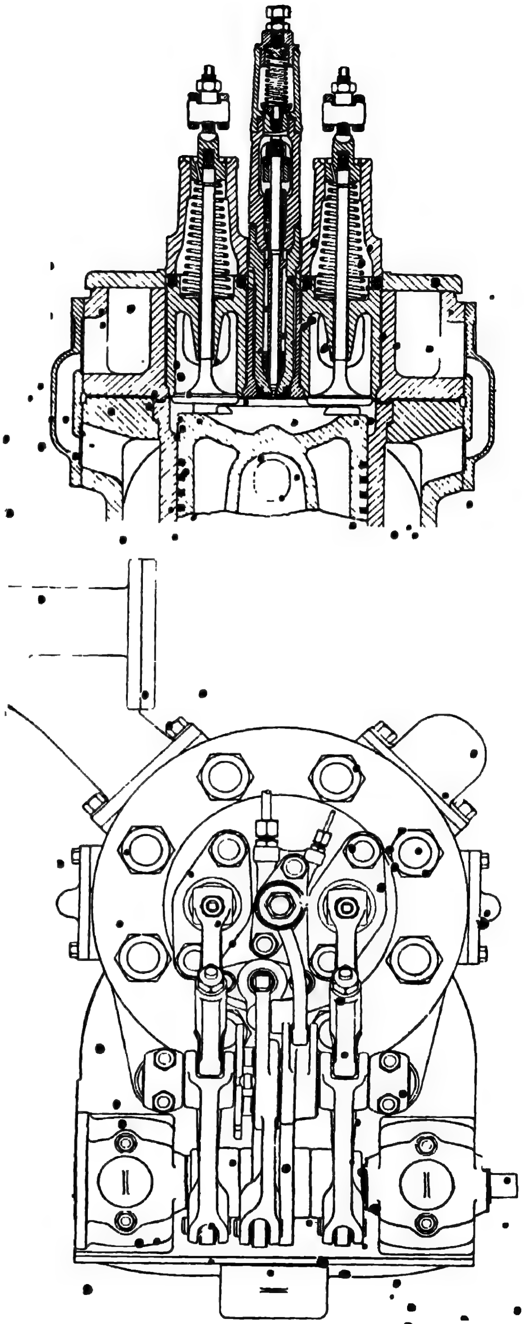

Fig. 1[illegible]4.—Mirrlees Engine Cylinder-head

The inlet and exhaust valves (fig. 185) themselves call for no particular comment. Both are of steel, and are provided with an ample radius under the head.

The fuel valve is shown in detail in fig. 186. The valve itself consists of a plain spindle provided with a conical seating, and held against its seating by means of a very stiff spring. Immediately surrounding the valve spindle is a light sleeve, which also rests on a coned seating, but the seating in this case is fluted in order to permit of the passage of fuel and air past it. Above the seating are a number of small perforated discs through

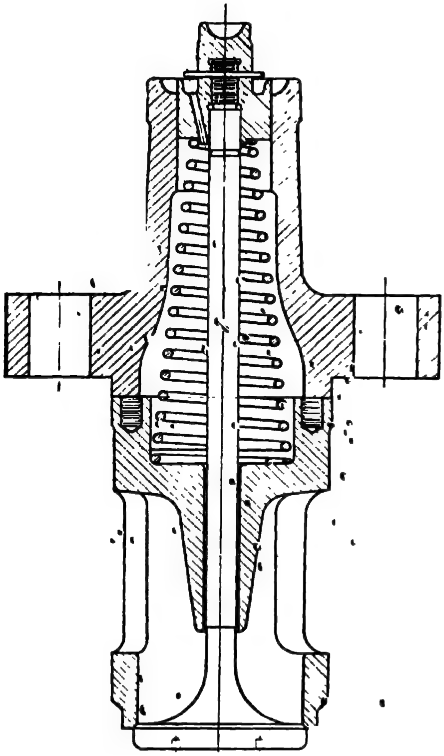

Fig. 185.—Mirrlees Engine Inlet and Exhaust Valves

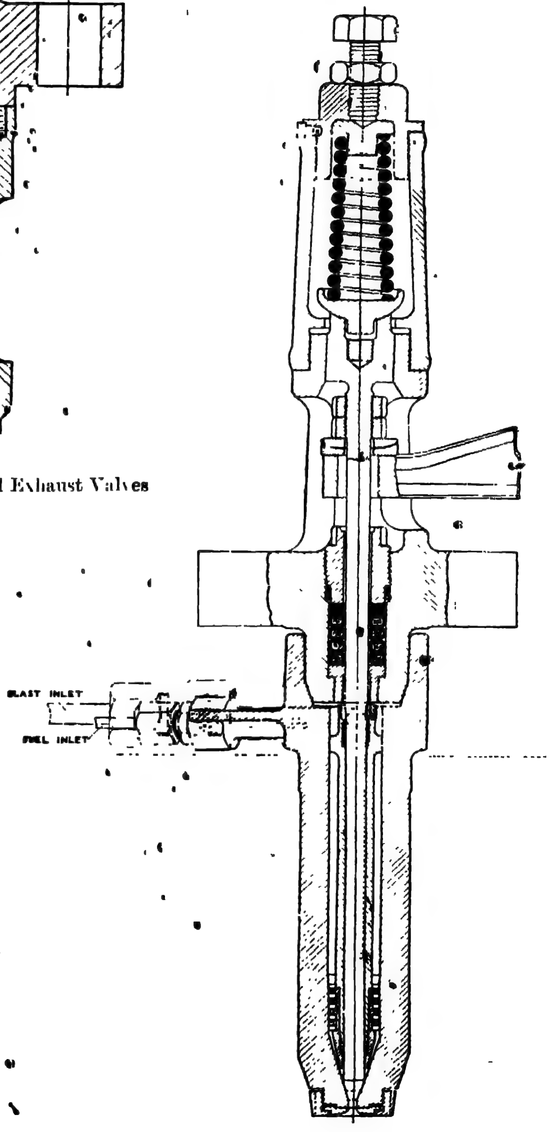

Fig. 186.—Mirrlees Engine Fuel Valve

which the fuel and air are driven. The fuel is delivered to the valve-chamber above these discs, and is driven through them by the highly compressed blast air when the valve is opened. The fuel valve, unlike the exhaust or inlet, opens outwards, and is operated by a large diameter cam mounted on the camshaft. The valve is timed to open a few degrees before the top dead centre, and remains open during about 10 per cent of the stroke.

The fuel pump is illustrated in fig. 187. It consists of an ordinary plunger pump, driven by an eccentric from the camshaft. The inlet valve of this pump is mechanically operated, and is under the control of the governor, which regulates the quantity of fuel by allowing the inlet valve to close later or earlier in the delivery stroke. The pump delivers fuel to the fuel valve slightly before it is required, so that the precise timing of the delivery is of no consequence. Two delivery valves are fitted, one above the other, and between the two there is a union and branch pipe by means of which the fuel can be returned to the fuel tank in order to stop the engine. It is clearly necessary to provide two valves, for, besides the increased reliability obtained thereby, the oil is delivered against the blast-air pressure, and, unless a second valve were fitted above the bypass, air would be driven back into the tank when the latter was opened.

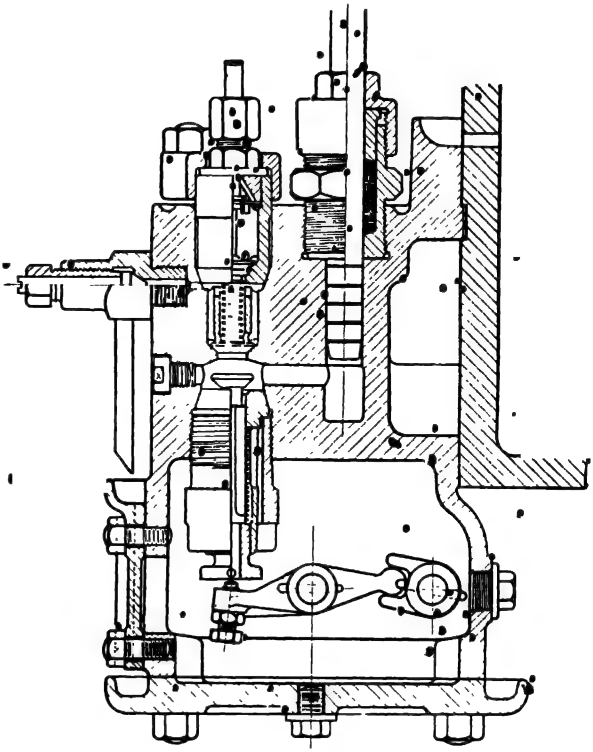

Fig. 187.—Mirrlees Engine Fuel Pump

The high-pressure air-compressor is arranged in two stages, in tandem. The lower-pressure stage compresses the air to a pressure of about 100 lb. per square inch, or, say, eight atmospheres, and

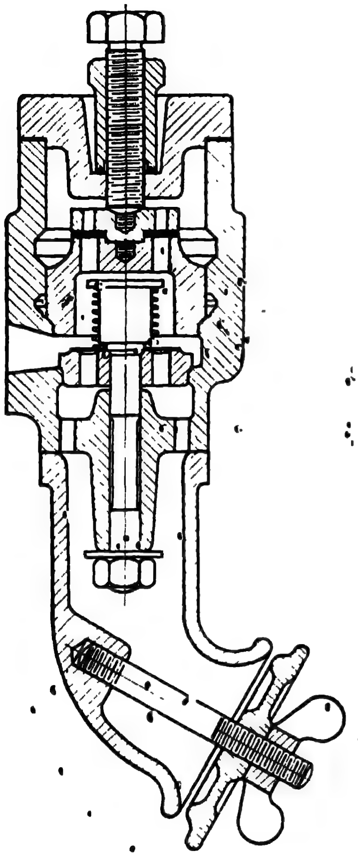

Fig. 188.—Compressor Valves, L.P. Stage

delivers it to the inter-cooler, which is bolted to one side of the A frame, and can be seen in section in fig. 181.

The second stage draws air from the inter-cooler, and compresses it to a pressure of about 900 lb. per square inch, or a further eight times. From the high-pressure air-compressor the air passes into an after-cooler which is incorporated with the inter-cooler, and from there to the storage bottles.

The valves used in the two stages of the air-compressor are shown in figs. 188 and 189; they consist of plain flat steel discs, oil-hardened and ground. No springs of any sort are used, except for the suction valve of the low-pressure stage, and only a very small lift is permitted. The governor is mounted on the vertical shaft above the spiral gears driving the camshaft. To ensure extreme sensitiveness, all the working parts are mounted on ball-bearings, and all the moving parts are enclosed in a dust-proof casing. The governor, in a Diesel engine, is only called upon to determine the timing of the suction valve of the fuel pump; consequently, very little power is required, and a comparatively small one can be employed. It is provided with a hand-adjusted "speeder" spring, by means of which the speed of the engine can be varied over a small range while running. The governor and its attendant gearing is shown in detail in fig. 190.

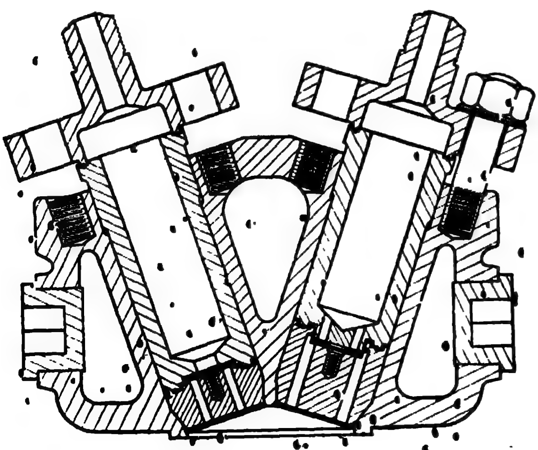

Fig. 189.—Compressor Valves, H.P. Stage

The leading dimensions of this engine are as follows:—

Bore	12 in.
Stroke	18·25 in
Piston area	113·1 sq. in.
Swept volume	1·192 cu. ft.
Speed	250 R.P.M.
Piston speed	760 ft. per minute.
Maximum power	52·5 B.H.P.
ηp (brake mean pressure)	80·7 lb. per square inch.
Compression ratio	13 : 1.
Air standard efficiency	64·16 per cent.
Diameter of valve ports	3·96 in.
Lift of valves	1·375 in.
Effective valve area	11 sq. in.
Ratio, piston area to effective valve area ...	10·3 : 1.
Weight of piston	364 lb.
Weight of reciprocating parts	530 lb.
Weight of reciprocating parts per square inch of piston area	4·69 lb.
Diameter of crank-pin	6·5 in.
Width of crank-pin bearing	6·96 in.
Projected area of crank-pin bearing ...	45 sq. in.
Ratio, piston area to projected area of crank-pin bearing	2·51 : 1.
Bore of compressor cylinder, L.P.	5·125 in.
Stroke of compressor cylinder, L.P.	8·5 in.
Swept volume of compressor cylinder, L.P.	175 cu. in.
Ratio, swept volume of cylinder to twice the swept volume of L.P. compressor ...	5·9 : 1.

The curves (figs. 191 and 192) show the fuel consumption, brake and indicated thermal efficiency, and the mechanical efficiency at various loads. The brake thermal efficiency can be measured directly, and with a very high degree of accuracy, but the mechanical efficiency is computed from the indicator diagrams, which are liable to an error of at least 5 per cent in the horse-power, and therefore nearly 20 per cent in the full-load mechanical efficiency. The curve illustrated, however, has been supplied by the makers, and represents the mean of a large number of tests, so that it may be regarded as fairly accurate in this case. From this curve it may be observed that on the normal full load of 50 B.H.P. the brake thermal efficiency is 33·2 per cent, and the mechanical efficiency 74 per cent. The indicated horse-power, as deduced from the indicator diagrams, is 71, and the indicated thermal efficiency is 45 per cent.

It is interesting to calculate the mechanical efficiency from the weight of the reciprocating parts and the area of the valves, and to see how this compares with the figure given by the makers. The

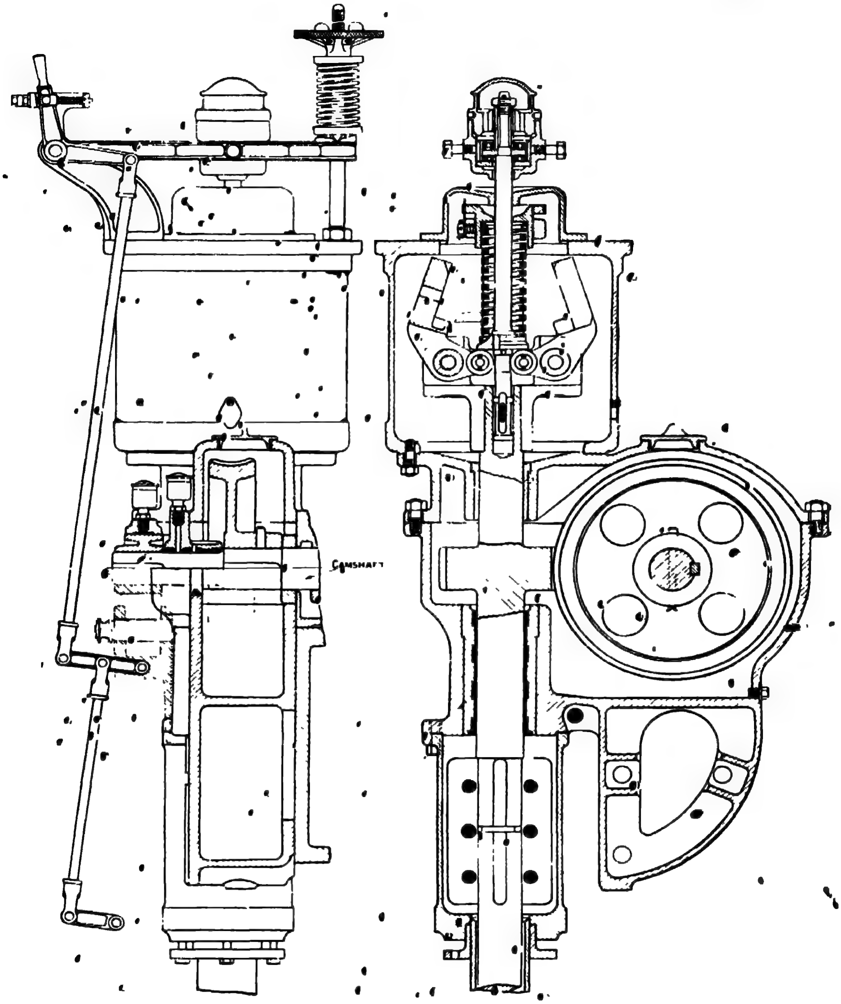

Fig. 190.—Section through Governor and Casing

area through the valves is 11 sq. in., and the ratio of piston to valve area is as 10·3 : 1, which corresponds to a velocity of

$$\frac{10 \cdot 3 \times 760}{60} = 130 \text{ feet per second.}$$

This is about the maximum velocity consistent with a reasonably high volumetric efficiency, and it indicates that with the valves

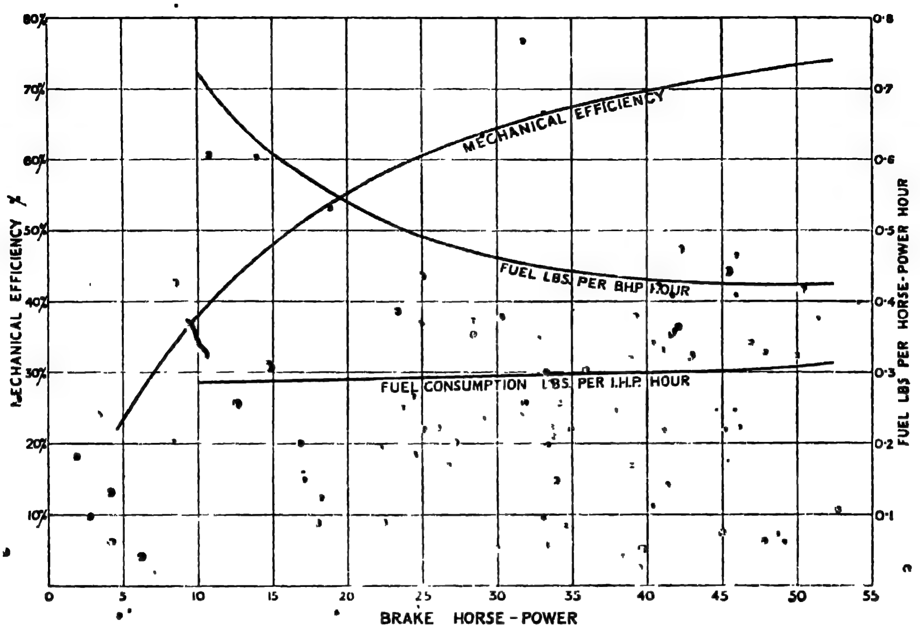

Fig. 191.—Curve showing Mechanical Efficiency and Fuel Consumption

arranged in the cylinder cover in this manner, it would not be possible to exceed a piston speed of 760 ft. per minute without a

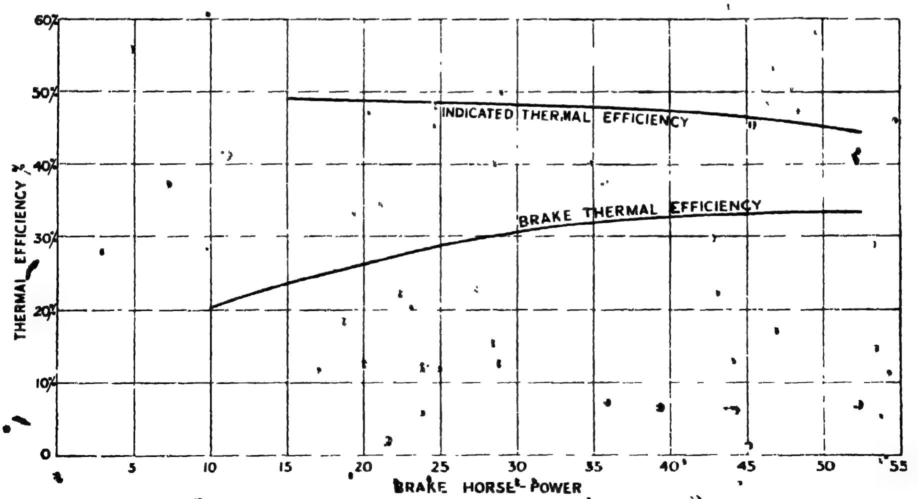

Fig. 192.—Curve showing Insulated Brake Thermal Efficiency at Various Loads

formidable increase in the fluid losses. A high inlet velocity is not necessary in a Diesel engine using air-blast injection, because the turbulence required to ensure rapid and complete combustion

is produced by the entry of the high-pressure blast-air. With this velocity the fluid or pumping losses will be in the neighbourhood of 3·5 lb. per square inch.

The piston friction in this case will be approximately 9·2 lb. per square inch.

The bearing and other friction may be put down as equivalent to nearly 3·5 lb. per square inch.

The high-pressure air-compressor may be regarded as having a mechanical efficiency of 85 per cent, and the indicated power absorbed in compressing the air is equal to a mean pressure of approximately 50 lb. per square inch, referred to the low-pressure cylinder and applied every revolution. It is therefore equal to a mean pressure of $\frac{50}{5 \cdot 9} = 8 \cdot 5$ lb. per square inch, when referred to the main engine piston, or 10 lb. per square inch when the mechanical losses of the compressor are taken into account.

The total fluid and friction losses now become:—

Fluid loss ...	...	...	3·5	lb. per square inch.	
Piston friction	...	...	9·2	„	„
Bearing friction	...	...	3·5	„	„
Air compressor	..	...	10·0	„	„
			26·2	„	„

At the normal load of 50 B.H.P. the brake mean pressure is 77 lb. per square inch.

The indicated mean pressure is therefore

$$77 + 26 \cdot 3 = 103 \cdot 3 \text{ lb. per square inch,}$$

and the mechanical efficiency

$$\frac{77}{103 \cdot 2} = 74 \cdot 5 \text{ per cent,}$$

which agrees fairly well with the figure supplied by the makers, and obtained from the indicator diagrams.

The mechanical efficiency appears to be somewhat lower than is usual in Diesel engines of this size, and the proportion of the power absorbed by the air-compressor is above the average.

At half-load the quantity of blast-air required for the fuel injection will be considerably reduced, but the other sources of loss will remain practically unaltered, except the piston friction, which may be reduced by about 10 per cent, due to the lower fluid pressures.

Reference to the curve of mechanical efficiency shows that at a load of 25 B.H.P. the mechanical efficiency is 60·5 per cent. The brake mean pressure corresponding to this load is 38·5 lb. per square inch, and the indicated mean pressure $\frac{100 \times 38{\cdot}5}{60{\cdot}5}$ = 60·6 lb. per square inch. The power absorbed in mechanical and fluid friction has therefore dropped from 26·2 lb. per square inch to 23·9 lb. per square inch; allowing for 1 lb. per square inch reduction in piston friction, the reduction in the blast-air amounts to 1·3 lb. per square inch. Such a deduction as this is interesting, rather as a check upon the accuracy of the mechanical efficiency curve than anything else. The reduction in the blast-air is just what one would expect, and leads to the conclusion that the mechanical efficiency, as deduced by the makers, must be very nearly correct. In fig. 193 are shown a number of indicator diagrams, taken from the engine under various loads; and in fig. 194, indicator diagrams taken from the two stages of the blast-air compressor.

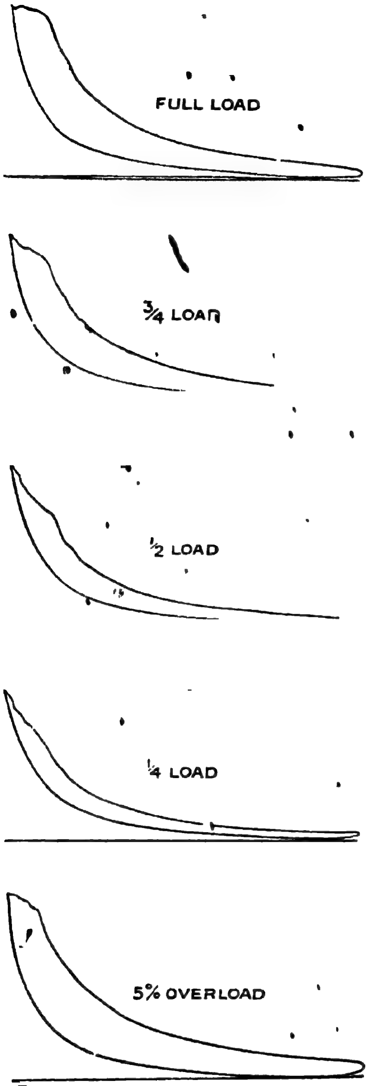

Fig. 193

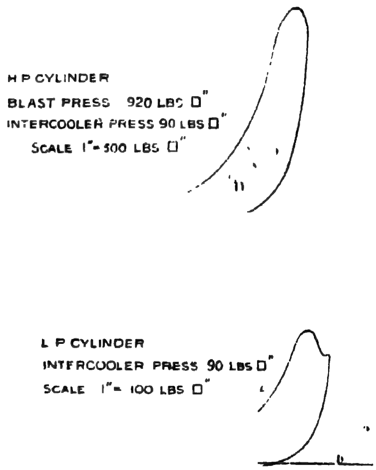

Fig. 194

Mirrlees Engine. Indicator Diagrams

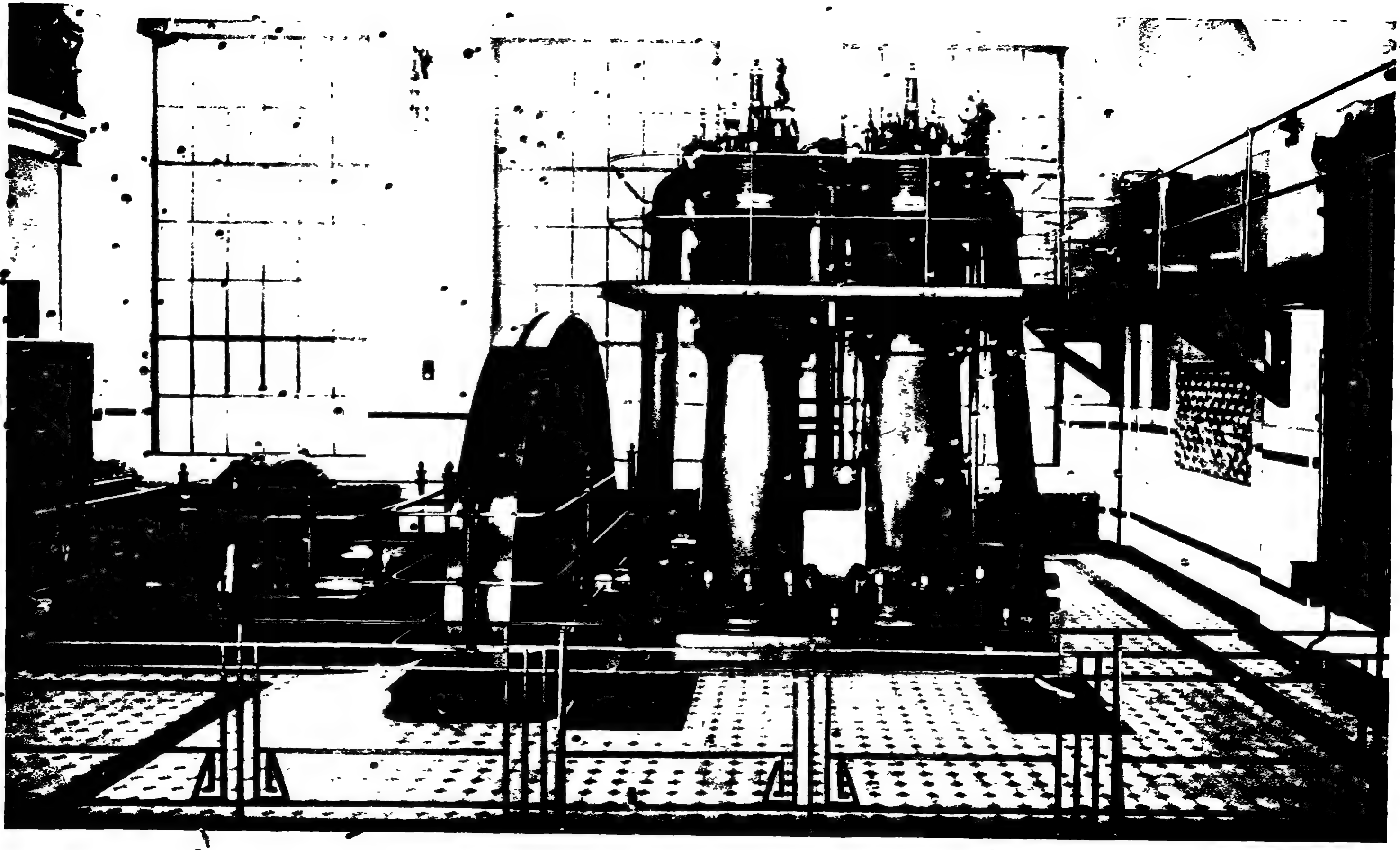

Fig 195.—70-H P. M A.N. Engine

The Mirrlees Diesel engine just described is typical of the whole class of slow-running stationary Diesel engines, which are built in sizes ranging from 8 to 200 B.H.P. per cylinder, by a large number of manufacturers both in this country and on the Continent. The design was originally evolved by the Maschinen Fabrik Augsburg-Nurnberg, and has since been followed by almost all the other makers of this class of engine, with the result that the difference between the various makes is so trifling as to be inconsiderable, and the detailed description of one such example as this will serve.

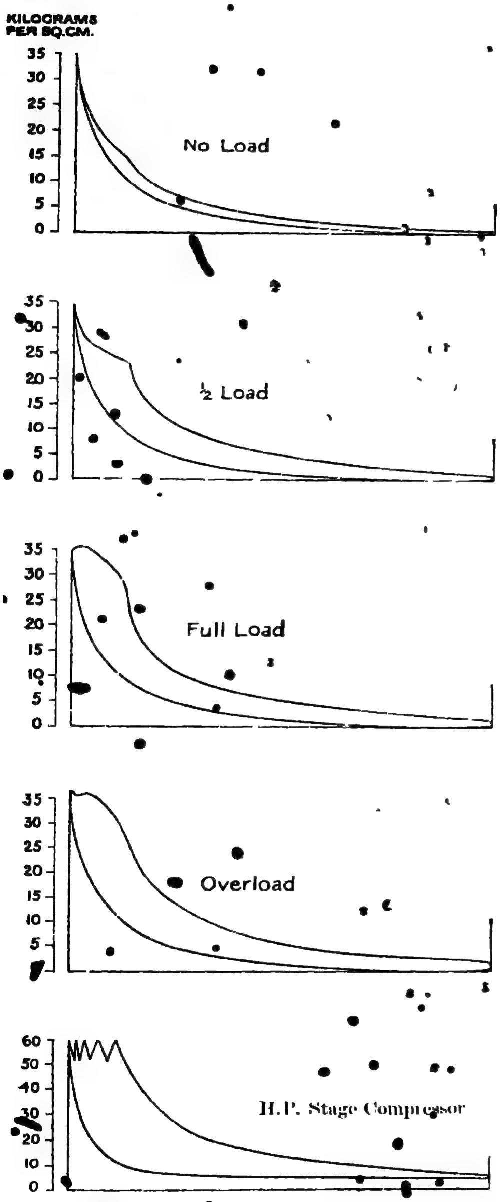

Fig. 196.—M.A.N. Engine. Indicator Diagrams

The M.A.N. Engine.—The Maschinen Fabrik Augsburg-Nurnberg—the original makers of the Diesel engine—now build an enormous number of these engines, of many different types, both four-cycle, two-cycle, high-speed, low-speed, vertical and horizontal. The M.A.N. vertical four-cycle engines of the low-speed type resemble the Mirrlees engine just described in all essential features, and it is not worth while to describe them in detail.

Fig. 195 is a typical example of one of the standard types built by this firm. The only important features in which this engine differs from the one last described are (1) in the disposition of the high-pressure air-compressor. In this case the compressor is mounted on the A frame

alongside the main cylinder, and is actuated by means of a rocking beam driven directly from the main piston, an arrangement which certainly makes for compactness. (2) The cylinder liner is supported upon a flange situated slightly below the top of the cylinder, in such a manner that there is a free circulation of water around it above this point; and the great thickness of uncooled metal, inevitable when the liner is carried in the ordinary manner, is thus avoided. This certainly seems a very desirable feature, especially in large powers, when the difficulty of adequately cooling this important part becomes a very serious one, owing to the great thickness of the metal.

(3) The cap of the connecting-rod big-end bearing is forged solid with the rest of the rod, and afterwards parted off for assembling and adjustment: this construction is expensive, but is eminently sound. As in the Mirrlees engine, the three main valves are arranged in a row across the head, with the result that the effective area of the inlet and exhaust valves is restricted, and that there is a wide belt of uncooled metal right across this very vital part of the cylinder-head. The engine illustrated has two cylinders, each of 300-mm. or 11·3-in. bore, and 460-mm. or 18·1-in. stroke, and develops 70 B.H.P. when running at 190 R.P.M., equivalent to a piston speed of only 573 ft. per minute. Tests on this engine have yielded the following results:

B.H.P.	I.H.P.	Mechanical Efficiency.	Fuel (lbs. per B.H.P. hour).	Indicated Thermal Efficiency	Brake Thermal Efficiency.	M.E.P. (lbs per sq. inch).	I.H.P. Compressor.	R.P.M.
		Per cent		Per cent.	Per cent.			
	24·1			47·7		25	0·9	196·7
-	26·2	-		45·5		27	0·98	196·5
40·8	63·6	65·7	0·458	15·0	28·9	66	1·36	193·3
68·0	88·6	78·7	0·410	42·4	32·5	93	2·11	191·4
81·3	100·6	83·2	0·413	39·7	32·3	106	2·26	189·9

The outstanding feature of these tests is the remarkably high mechanical efficiency obtained: this is clearly due to the unusually small amount of power absorbed by the air-compressor. No statement is made as to the nature of the fuel used, but it is probable that it was some very light oil which required little air for pulverization. A series of indicator diagrams obtained during this test are shown in fig. 196. From this diagram it will be observed that the compression pressure employed is approximately 500 lb. per square inch, and also that the admission of fuel is irregular, and takes place too late in the stroke. On the lighter loads a certain proportion of

the fuel enters while the piston is passing over the top dead centre, but the main bulk of it is not admitted until so late in the stroke that it has the effect rather of producing a hump in the expansion curve than a flat top to the diagram. It may be that this is due to the employment of an insufficient supply of blast-air.

The following test figures were obtained from a M.A.N. Diesel engine having four cylinders, each 480-mm. or 18·9-in. bore, 680-mm. or 26·8-in. stroke, and developing 480 B.H.P. at 170 R.P.M. They are of considerable interest, because during this test the fuel used was tar oil, which has a high ignition point, and consequently requires a very high temperature, and particularly fine pulverization. In order to ensure ignition it is usual to admit a small quantity of light crude oil ahead of the tar oil. This light oil burns immediately, and so raises the temperature of the air before the tar oil is admitted. In the test figures given below, an ignition charge consisting of about 10 per cent of light gas oil was admitted on the ½- and ¾-load trials, but not on the full load, the high temperature of the piston-head in this case rendering the use of an ignition charge unnecessary.

B.H.P.	I.H.P.	R.P.M.	Mechanical Efficiency.	M.E.P.	I.H.P. Compressor.	Indicated Thermal Efficiency.	Brake Thermal Efficiency.	Blast-air Pressure (lbs. per sq. inch).
			Per cent.			Per cent.		
485·2	668·1	167·6	72·5	106	33·5	44·4	32·2	820
485·6	668·5	167·7	72·5	106	33·3	43·5	31·5	820
371·1	539·5	170·4	68·8	83·5	28·5	45·2	31·1	717
254·4	422·7	171·8	60·3	64·5	25·7	47·4	23·6	623
253·6	423·5	172·1	59·7	64·5	26·6	47·3	23·2	620

Indicator diagrams taken during these tests are illustrated in fig. 197, and show very fair adjustment of the fuel valves, though combustion is somewhat retarded at ½- and ¾-load.

A further series of tests taken from the same engine, but using a different fuel, yielded the following results:—

B.H.P.	I.H.P.	R.P.M.	Mechanical Efficiency.	M.E.P.	I.H.P. Compressor.	Indicated Thermal Efficiency.	Brake Thermal Efficiency.	Blast-air Pressure (lbs. per sq. inch).
			Per cent.			Per cent.	Per cent.	
500·9	690·6	167·4	72·5	110	33·1	44·7	32·4	826
486·0	680·0	167·5	71·5	108	31·4	45·0	32·1	805
374·0	546·9	170·1	68·4	86	30·5	48·9	33·4	730
370·2	547·4	170·7	67·5	86	31·1	49·4	33·3	735
259·4	430·7	172·2	60·4	68	29·0	50·0	30·1	680

In this second series of tests, both the brake and indicated

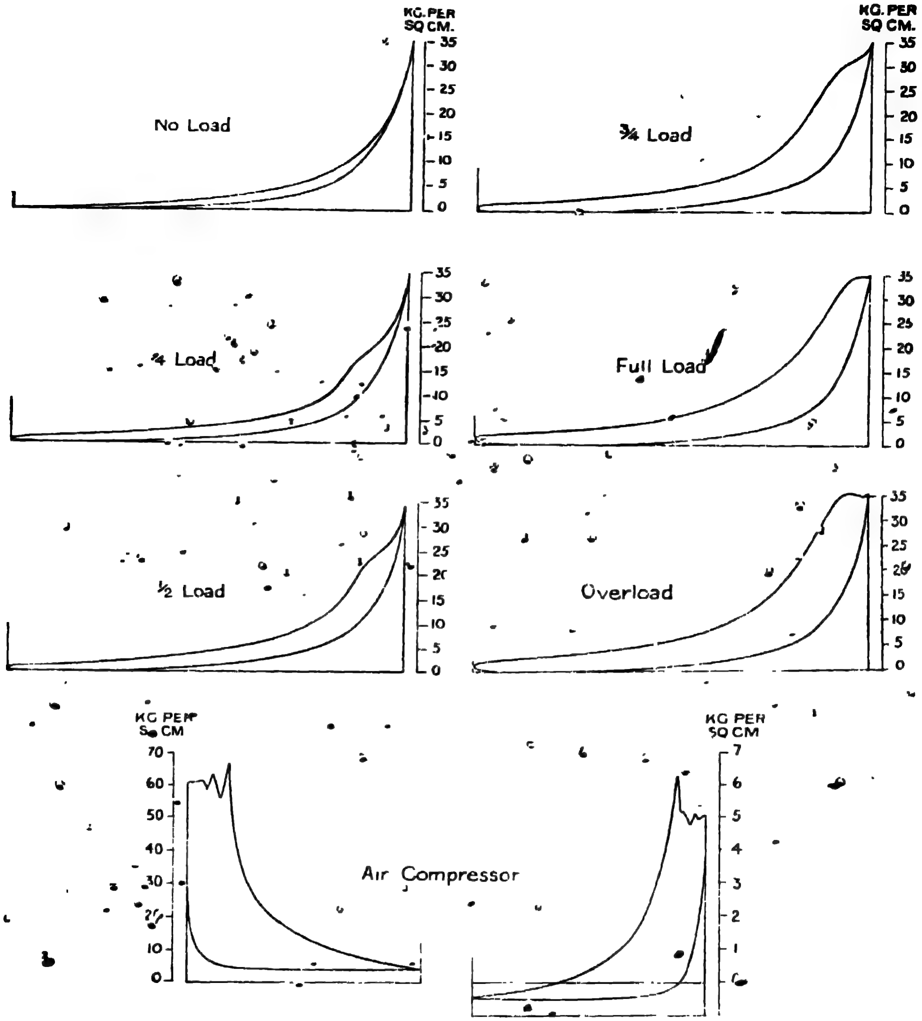

Fig. 197.—Diagrams from 480-H.P. M.A.N. Engine

thermal efficiencies are higher on the lighter loads. This may, of course, be due to the different fuel employed, but it is also probably due to some extent to the higher blast-air pressure employed on the lighter loads, which will certainly tend to increase the apparent indicated thermal efficiency. If, in the last line, the indicated horse-power absorbed by the air-compressor be deducted from the total indicated power, the indicated thermal efficiency then becomes 46·4 per cent, which is much nearer the true figure.

Tabulating the results of the second test with the I.H.P. of the blast-air compressor deducted from the total I.H.P. gives the following results:—

B.H.P.	I.H.P.	M.E.P.	Indicated Thermal Efficiency.
500·7	657·5	104	43·5 per cent.
486	648·6	101	42·8 „
374	516·4	82	46 „
370·2	516·3	82	46·5 „
259·2	401·7	63·2	46·6 „

It is, of course, necessary to deduct the indicated power of the air-compressor before any idea can be obtained as to the relative efficiency, or any just comparison made between the indicated thermal efficiencies of Diesel and other engines in which compressed air is not admitted to the cylinder during the expansion stroke. Even though the indicated power of the air-compressor be deducted, it is still not altogether fair to compare the results with those obtained from other engines, but the discrepancy is then comparatively small. In this engine the air-standard efficiency regarded as a constant-volume engine is 65 per cent, and the relative efficiency on half load is, approximately, 72 per cent.

Both the above sets of tests were very complete, and were carried out with a high degree of accuracy. Data are given as to the heat carried away by the cooling-water and the exhaust, and also the temperature of the gases in the exhaust pipe. These figures have been omitted because (1) the heat carried away by the cooling-water contains a large and indeterminate percentage which should be debited to the exhaust, and does not give any clue as to the amount of heat lost during combustion and expansion; (2) the temperature of the exhaust gases was measured after they had expanded to atmospheric pressure, and yielded up a certain proportion of their heat to the jacket-cooling water. Hence it gives no clue as to the real temperature of the gases in the cylinder at the end of the expansion stroke, which is the chief point of interest and importance.

The American Sulzer.—The Busch-Sulzer engine, illustrated in fig. 198, is an American production, and is built by the Busch-Sulzer Diesel Engine Company of St. Louis, U.S.A. This engine differs substantially from what may be described as the standard Augsburg design, and is modelled rather upon the Westinghouse vertical gas-engine. It is built in four sizes, rated at 75, 120, 170, and 225 B.H.P., three cylinders being used in each case. It is a little difficult to understand why three cylinders should have been adopted as the standard, for three-cylinder engines are liable to considerable vibration, owing to the large unbalanced fore and aft

couple, which causes the whole engine to "pitch" unless exceptionally heavy foundations are provided to counteract this tendency.

Dealing with the mechanical features, the crank-chamber is

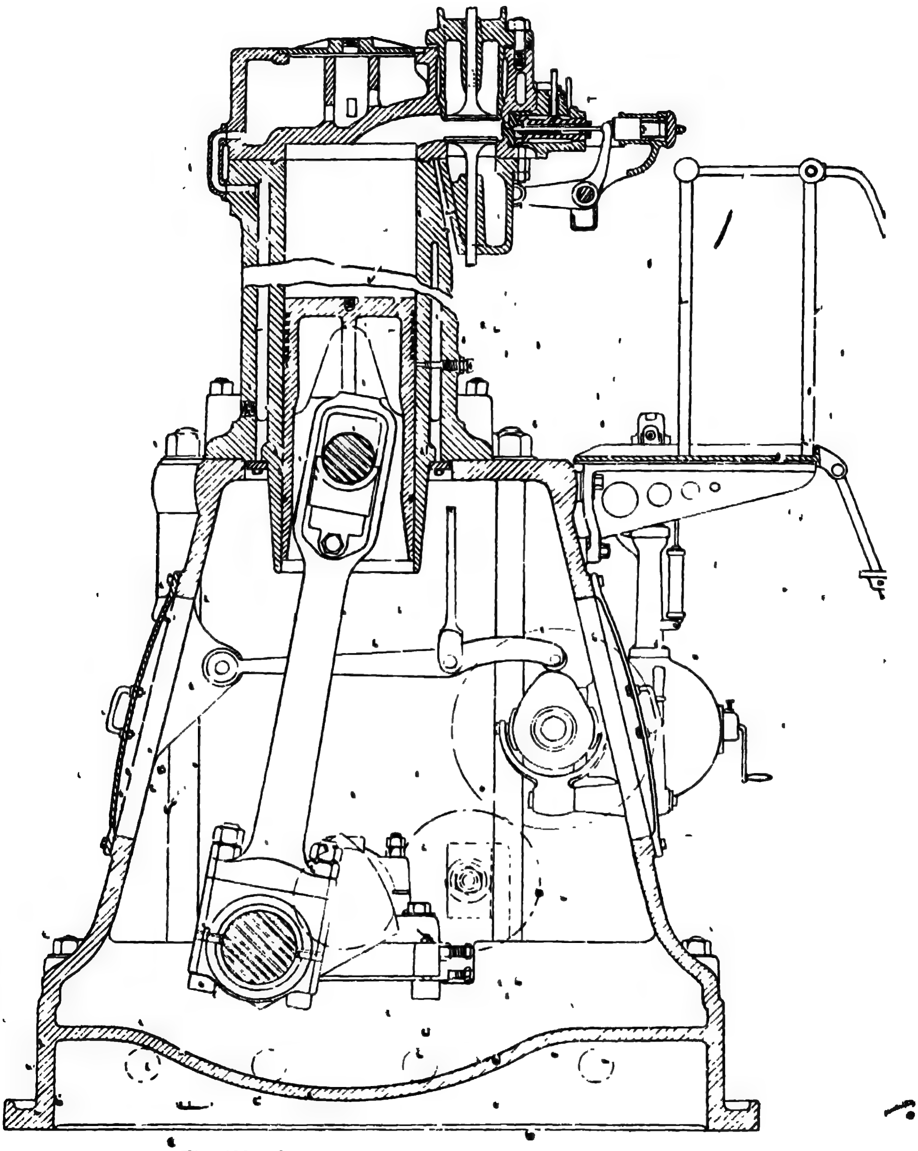

Fig. 108.—Sectional View of Busch-Sulzer Diesel Oil-engine

totally enclosed, and is made separate from the cylinder. It is strengthened by means of vertical steel bolts, which, however, are not carried up to the top of the cylinder, and do not relieve the cylinder barrel from the direct tensile stresses. No separate cylinder

liner is employed, but in order to prevent distortion and stresses due to the unequal expansion of the inner and outer walls of the cylinder, the lower end of the water-jacket is left open. It is afterwards closed by a light sealing ring, so that the inner and outer walls are free to expand independently. The valves in this engine are not fitted in the cylinder-head but in a side pocket, as is usual in gas-engine practice. It has already been pointed out that in a Diesel engine it is a matter of great importance that the combustion chamber shall be as compact and symmetrical as possible, in order that the pulverized oil may be thoroughly distributed throughout the whole bulk of the air. This construction has the advantage that it does not interfere with the efficient water-cooling of the cylinder-head; but, on the other hand, it does interfere seriously with the efficient cooling of the cylinder barrel in the neighbourhood of the exhaust valves, and just at a point where any distortion might lead to excessive piston friction, and possibly to seizing of the piston.

The high-pressure air-compressor is entirely independent of the main engine, and is driven by either a separate motor or by belt.

The following tests have been carried out by Drs. H. W. Harper and L. R. Bailey, of the University of Texas, upon a three-cylinder 225 B.H.P. engine installed at the Hugo Ice and Light Company, Oklahoma, U.S.A. The high-pressure air-compressor in this case was driven by an electric motor, supplied with current generated by the main engine, and the quantity of current used for this purpose has been deducted from the total output. Thus the losses in the electric motor, as well as the power absorbed by the air-compressor, are debited against the main engine, which is hardly fair. The author has endeavoured to correct this as far as possible, on the assumption that the efficiency of the motor is approximately 85 per cent. With this correction the results obtained are as follows:—

B.H.P.	R.P.M.	Brake Thermal Efficiency.	B.H.P. of Air Compressor.	Blast-air Pressure (lbs. per sq. inch).
		Per cent		
245·5	161·9	30·5	13·5	1140
221·5	165	30·8	11·5	970
163·8	164	29·8	10·2	870
113·7	167	28·5	10·3	830
51·3	169	18	7·9	670

No indicator cards were taken during the test, and no attempt was made to ascertain the mechanical efficiency, nor are the dimensions

of the cylinder given, so that it is not possible to analyse these results. Compared with European engines the brake thermal efficiency is low, which is just what might be expected as the result of the use of a pocket. The fuel used during this test was Oklahoma crude oil, which has a calorific value of 18,986 B.T.U. per pound, and a specific gravity of only 0·8531: that is to say, a very light oil and one requiring very little air for pulverization, and it is not, therefore, very easy to understand why so high a blast-air pressure was required.

High-speed Engines.—For auxiliary purposes on board ship, and for "stand-by" purposes, the high-speed type of four-cycle Diesel engine is generally employed, because its first cost, bulk, and weight are all lower. It is, however, a less efficient engine, because the mechanical efficiency is generally lower owing to the weight of the reciprocating parts, and also owing to the fact that, in order to obtain sufficiently rapid combustion, a greater quantity of blast-air must be employed. Most of the makers of slow-speed four-cycle engines manufacture also the high-speed type, generally in comparatively small sizes, and with a comparatively large number of cylinders, in order to reduce the weight and bulk to the lowest possible limit. Such engines are also used for the propulsion of submarines, and for this purpose either six or eight cylinders are employed, the power ranging from 500 to 1000 B.H.P. per engine. Two such engines are usually fitted, driving twin screws.

The high-speed type of Diesel engine is almost invariably built with a totally enclosed crankcase, and employs forced lubrication to all bearings, under a pressure of from 50 to 60 lb. per square inch. The construction of the cylinders and cylinder-heads is the same as in the low-speed type, but cast steel is frequently used, both for the cylinder-heads and bodies, though the actual cylinder liners are invariably of cast iron. The valves are arranged in a row across the cylinder-head, as in the slow-speed type, but the fuel valve is, in some cases, placed slightly out of centre, in order to allow of larger inlet and exhaust valves, and to provide a circulation of water between them. From a mechanical point of view this is distinctly an improvement, but the distribution of fuel throughout the combustion space is not so thorough as when the fuel valve is central. Although these engines employ a comparatively high rotative speed, the actual piston speed is little or no higher than in the slow-running type. Consequently the ratio of valve area to piston area can remain approximately the same. The stroke is generally very short, the

stroke-bore ratio being usually only about 1·1 : 1. With a compression ratio of 13 : 1, this means that the depth of the compression space is less than one-tenth of the diameter, a proportion which is most unfavourable, both on the grounds of the even distribution of the fuel and also of heat loss to the cylinder walls. A short stroke is, however, a necessary evil, if the cost and weight are to be reduced to the lowest limit.

Apart from the question of rapid combustion, it is not possible to run a Diesel engine of the ordinary four-cycle type at a high piston speed, because there is not room in the cylinder-head for large enough valves. The cylinder-head has to accommodate no less than four valves, each with detachable seatings, and it is no easy matter to accommodate these valves together with their ports and passages, in the very restricted space available, and at the same time provide for adequate water-cooling. In practice, the piston speed is limited to about 800 ft. per minute, and even at this speed it is almost impossible to obtain a volumetric efficiency of much over 70 per cent. It is clear that if larger valves could be fitted and a higher volumetric efficiency obtained, higher mean pressures could be employed without loss of efficiency; or, conversely, the same mean pressures could be employed with lower maximum temperatures and a higher efficiency.

Owing to the high temperature necessary to produce a high mean pressure with a comparatively low volumetric efficiency, it is often found desirable to cool the pistons even in relatively small engines. Water-cooling of the pistons is by no means an easy matter in any engine, and is especially difficult in a high-speed enclosed type, for the water must be supplied to the pistons at a very high pressure to ensure against water-hammer; consequently, leakage is almost unavoidable. In a totally enclosed engine, leakage of water is particularly objectionable, for it, of course, mixes with the lubricating oil, and seriously impairs the lubrication of the bearings. To obviate these difficulties and dangers, oil-cooling is sometimes employed. The lubricating oil is circulated through the pistons, or, in some cases, a jet of oil is directed against the underside of the piston-head. The oil then returns to the base-chamber, and is circulated through an oil-cooler of large capacity in order to remove the heat taken up from the pistons. This arrangement is satisfactory for comparatively small engines, but ordinary lubricating oil is a poor conductor of heat, and carbonizes very readily, so that it is not suitable for large pistons.

Deutz Horizontal Engine. — Messrs. The Gasmotoren-Fabrik Deutz, of Cologne, have recently brought out a design of four-cycle engine which, in all essential features, resembles the horizontal gas-engines built by that firm. Several models of Diesel engines are produced by the Deutz Company, ranging from 12 to 40 horse-power per cylinder, and in all cases provision is made for running on tar oils with the addition of a small quantity of ignition oil. The arrangement of the valves and the shape of the combustion chamber are identical with the usual gas-engine construction,

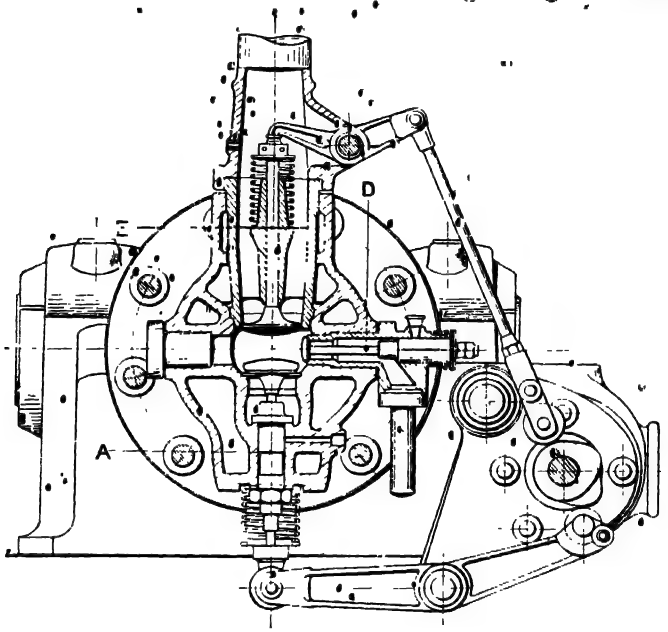

Fig. 199.—Deutz Horizontal Engine

as shown in the cross-section, fig. 199. The peculiar points about this engine lie in the fuel valve and the high-pressure air-service. The fuel valve is of the open type, and, as has already been explained, the ignition oil is admitted to the passage behind the flame-plate quite early in the compression stroke, where it vaporizes, the vapour passing back into the main passage through a small communicating port immediately at the back of the flame-plate. As soon as the main air valve is opened, the ignition-oil vapour enters the cylinder and is immediately burnt at constant volume, thus causing a sudden rise of pressure, up to about 750 lb. per square inch, and a corresponding rise of temperature sufficient to ignite the tar oil.

Although the ignition oil is present in the cylinder, or at least

in a passage communicating with the cylinder, throughout the compression stroke, yet there is no danger from pre-ignition, because even if the vapour should find its way into the cylinder-head before the end of the stroke, the quantity is so small that it would not raise the pressure of the air more than about 50 per cent, and could have no serious consequence. The tar oil is delivered direct to the fuel-valve seating by means of a cam-operated pump, which is timed so that the stroke of the pump coincides with the opening of the needle-valve. The latter is placed at right angles to the spray passage, so that it can be operated direct from the side shaft. The high-pressure blast-air compressor is of the two-stage tandem type: it is mounted alongside one of the main bearings, and driven by means of a crank-pin fitted to the end of the main shaft. No air storage is provided, but the high-pressure air passes directly from the second stage of the compressor to the fuel valve. The regulation of the blast-air, therefore, takes place automatically, for the quantity delivered per stroke remains the same at all loads, but the pressure varies according to the quantity of tar oil delivered by the fuel pump. The greater the quantity of fuel the greater the resistance through the fuel-valve and flame-plate, and therefore the greater the air pressure. For starting, the ordinary air-starting valve is fitted, the necessary air being stored in steel bottles at a pressure of about 175 lb. per square inch. The supply of air for this purpose is taken from the inter-cooler fitted between the high- and low-pressure stages of the compressor. The whole arrangement is extremely neat and well-designed. The inter-cooler is made concentric with the air-compressor cylinder, and incorporated in the same jacket. Independent tests carried out on a 20-horse-power Deutz engine, using Galician gas oil of about 18,500 B.T.U. per pound, yielded the following results:—

Load.	Fuel Consumption (lbs. per B.H.P. hour).	Brake Thermal Efficiency.
24 B.H.P.	0·46	30 per cent.
20·7 „	0·453	30·4 „
8·9 „	0·557	24·7 „

No further particulars of this test are available, but in view of the fact that the maximum pressure rose to over 750 lb. per square inch the thermal efficiency is rather disappointing.

Double-acting M.A.N. Engine.—At their Augsburg works the M.A.N. Company have recently undertaken the manufacture of double-acting four-cycle Diesel engines, in powers up to 500-horse-

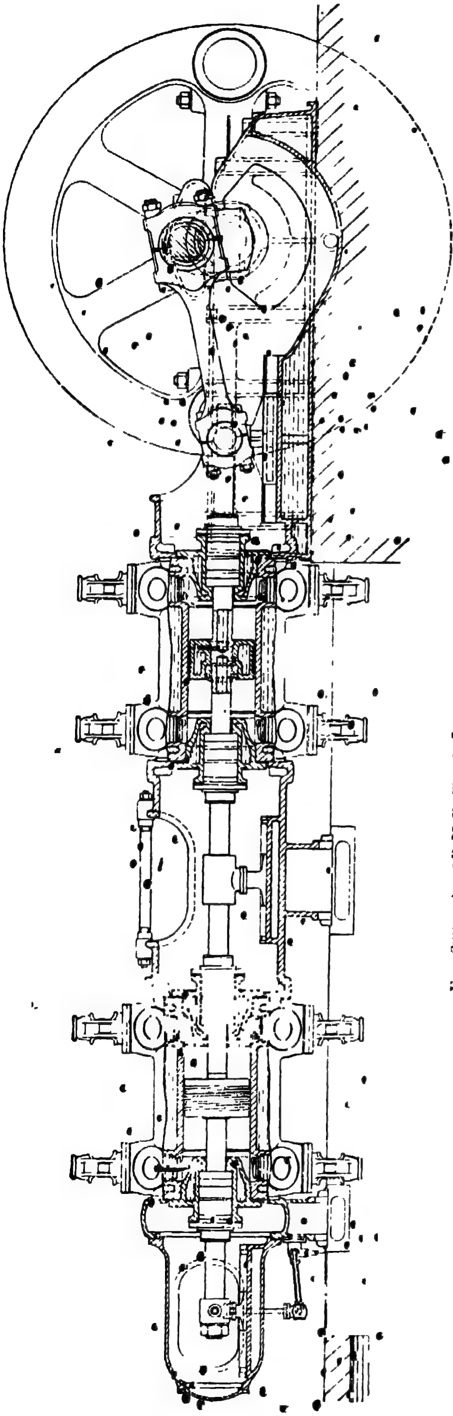

Fig. 200. 2000 B.H.P. Tandem Double-acting M.A.N. Diesel Engine.

power per cylinder, the largest engines being of the four-cylinder twin-tandem type, and developing about 2000 B.H.P. In this engine, of which sections are shown in figs. 200 and 201, the usual gas-engine practice has been followed throughout. The only features of any special interest are to be found in the combustion chambers. In this case the valves open into nearly spherical pockets, which together form the whole of the combustion space, for the clearance between the piston and cylinder covers is reduced to the lowest possible limit. Each of these separate combustion chambers is provided with an independent fuel valve. In this manner the problem of the unsymmetrical form of the combustion chamber in a double-acting engine is to some extent overcome, and the piston-rod is shielded from the direct blast of the burning oil; but it must be remembered that from the point of view of efficiency each cylinder end must be regarded as two small separate cylinders, each, however, of a very efficient form. This is certainly a simple solution of the problem, but it is one that seems likely to lead to trouble from cracked cylinders, because

the heat-flow to the metal surrounding the exhaust valve is more intense than at any other point in the cylinder, and if the same part be also used as a combustion chamber the rate of heat-flow may be so intense as to cause failure. In any case it seems somewhat undesirable to add to the rate of heat-flow in this severely stressed portion of the cylinder.

In the large experimental two-cycle double-acting engine built by Messrs. Krupp the combustion chamber is in the form of an annular ring, and this is accomplished by so forming the piston that the central portion almost touches the cylinder cover; but the outer portion is cut away, leaving an annular space between the piston and cover. The fuel valves in this case are probably fitted tangentially in order to produce a slight whirling motion around this chamber, and so assist in the distribution of the fuel. The piston-rod packing in the Augsburg double-acting engine does not call for any particular comment, for it is similar to that used in gas-engine practice. It is obvious that the very greatest attention must be paid to this to ensure against leakage, and the danger which leakage involves in any Diesel engine. Up to the present time only a very limited number of double-acting engines have been turned out by the Augsburg works, and they must still be regarded as being in somewhat of an experimental stage, although a few such engines have recently been put into actual service. No particulars are available as to tests made on these engines.

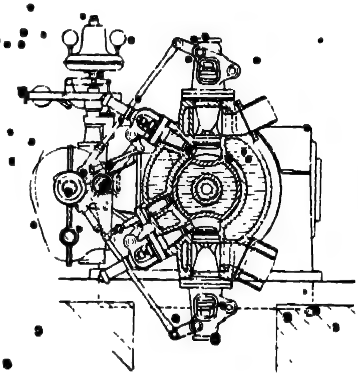

Fig. 201.—Cylinder Head

Double-acting Diesel engines have also been built experimentally by both Messrs. Krupp and the Nurnberg branch of the M.A.N. Company, but in both cases the two-stroke cycle has been adopted. Both these firms have had large engines, developing about 2000 B.H.P. per cylinder, on the test-beds for two or three years, and great secrecy has been maintained as to the results achieved. There is, however, good reason to believe that the experiments have been disappointing. The large Nurnberg engine was almost completely wrecked, and a number of men killed and injured by, it is said, an explosion of oil vapour in the scavenge air trunk.

CHAPTER XXX

TWO-CYCLE DIESEL ENGINES

The two-cycle Diesel engine of the slow-speed, stationary type has recently been developed for very large powers. The limit of power commercially obtainable from the four-stroke single-acting type has so far been about 250 to 300 B.H.P. per cylinder. This limit is set by the thickness of metal in the cylinder walls and combustion head. The thickness cannot be increased beyond a certain point, because the difference of temperature between the internal and external walls of the cylinder becomes so great that the stresses set up by unequal expansion neutralize any advantage gained by further thickening of the walls. Also the temperature of the inside surfaces becomes so high that there is great difficulty in adequately lubricating the pistons. The actual thickness of the walls is, of course, governed by the maximum abnormal pressures, and since these are the same in all Diesel engines it follows that it is proportional to the diameter of the cylinder. In the case of two-cycle Diesel engines it is clear that nearly double the power can be obtained from a given size of cylinder and, therefore, from a given thickness of metal; but in this case, owing to the greater number of expansion strokes, the heat-flow is more rapid and the temperature difference greater. In practice, therefore, other things being equal, the power obtainable from a single cylinder is not double that from a four-cycle engine, but is about 60 per cent greater. There is, however, this advantage, which applies to many types of two-cycle engines: the cylinder-head need not be pierced to accommodate a large number of valves, and therefore weakened structurally and imperfectly cooled. In those two-cycle engines in which bottom-scavenging is employed, it is necessary to provide only for the fuel valve, and in most cases the air-starting valves, in the cylinder-head. These are both small valves, and do not weaken the head or interfere with the cooling to any serious degree. Hence the metal can be made somewhat thinner, or conversely a larger diameter can be safely employed for the same limiting thickness of metal.

While the four-cycle Diesel engine has now settled down to a practically stereotyped design, the two-cycle engine is still very much in the experimental stage, and designers are by no means unanimous as to the best system of scavenging, which is the essential feature in all two-cycle engines. The problem of scavenging is, of course, very simple as compared with the gas-engine, because air alone is employed, and there is not the same necessity for encouraging stratification and guarding against loss through the exhaust ports. In some cases the scavenging air is admitted through valves in the cylinder-head, in others through ports uncovered by the piston, either with or without a delaying valve, and in others again both by inlet valves in the head and ports in the lower parts of the cylinder. The latter seems to be simply a blind attempt to force air into the cylinder at any cost.

All things considered, it would appear that for large engines the best system is probably bottom scavenging, with the addition of a delaying valve. This system has only recently been applied to Diesel engines by Messrs. Sulzer Brothers, of Winterthur. It has the great advantage that it provides ample port area; it leaves the cylinder-head free for the thorough circulation of the cooling-water, and does not weaken it structurally. In addition to the above, the method of scavenging with two opposed pistons, as in the Oechelhauser gas-engine, has been very strongly advocated by Professor Junkers, in Germany, and has been adopted by several manufacturers for marine engines, and also by the Allgemeine Elektricitäts Gesellschaft, of Berlin, for stationary engines, though the latter firm build only comparatively small engines designed to run at a high speed.

Sulzer Two-cycle Engine.—The engine illustrated in fig. 202 is built by Messrs. Sulzer Brothers, of Winterthur, and develops 2400 B.H.P. when running at a speed of 150 R.P.M. In this engine, which is not of the latest type, scavenging is effected through valves in the cylinder-head, and four valves are employed. These valves are yoked together in pairs, and operated by means of two rocking levers. Owing to their great size, and the very rapid opening and closing which is necessary in a two-cycle engine, they are, as might be expected, somewhat noisy in operation. The combustion-head is so arranged that the intervening space between each valve is cored out for water circulation, and the four valve ports lead into a common chamber surrounding the head. The fuel valve is fitted centrally in the cylinder-head, and the air-starting valve

between one pair of scavenge valves. Exhaust ports are provided all around the circumference of the liner, and the exhaust gases pass first into an annular chamber cast in the cylinder casting. Thence they pass into the water-jacketed exhaust pipes, of which a separate one is employed for each cylinder—a very desirable feature, especially in two-cycle engines, which are greatly affected by the pulsation set up in a common exhaust pipe.

Between the exhaust ports the liner is considerably thickened, in order to enable holes to be drilled through it parallel to the bore, for the circulation of cooling-water. The whole of the water is admitted below the exhaust belt, and passes upwards through these holes in the liner, thus ensuring thorough cooling of the bars between the ports.

The upper part of each piston is partitioned off, and water is circulated through it under a considerable pressure by means of telescopic pipes.

For the scavenging air, two separate pumps are provided, each being double-acting, and the admission and delivery is controlled by a piston valve, operated from an eccentric on the crankshaft. The use of mechanically operated delivery valves, as in this instance, is only applicable to engines running at a constant speed, in which the exhaust back-pressure remains approximately uniform. Even so, however, the valve setting cannot be right for both full load and light load, though it is very doubtful whether the extreme variation in load, in a constant-speed engine, is sufficient to justify the use of automatic valves, with the reduced area and greater resistance which they offer. The cross-heads of the scavenge pumps are employed as pistons for the low-pressure stage of the blast-air pumps, the high and intermediate stages being operated by means of rocking levers actuated from the cross-heads.

For the lower-pressure stages of the blast-air compressor, Gutermuth valves are employed. These valves consist of a spiral of thin, hard phosphor-bronze strip, the end of the spiral being brought out tangentially and acting as a plain flap-valve. By the use of a long spiral the bending stress is distributed throughout a large area, and not localized at any one point. In these large engines means are provided both for controlling the quantity of blast-air and the timing and opening of the fuel valves, according to the load. For this purpose a balanced throttle is fitted in the suction pipe of the low-pressure stage of the compressor, which is actuated

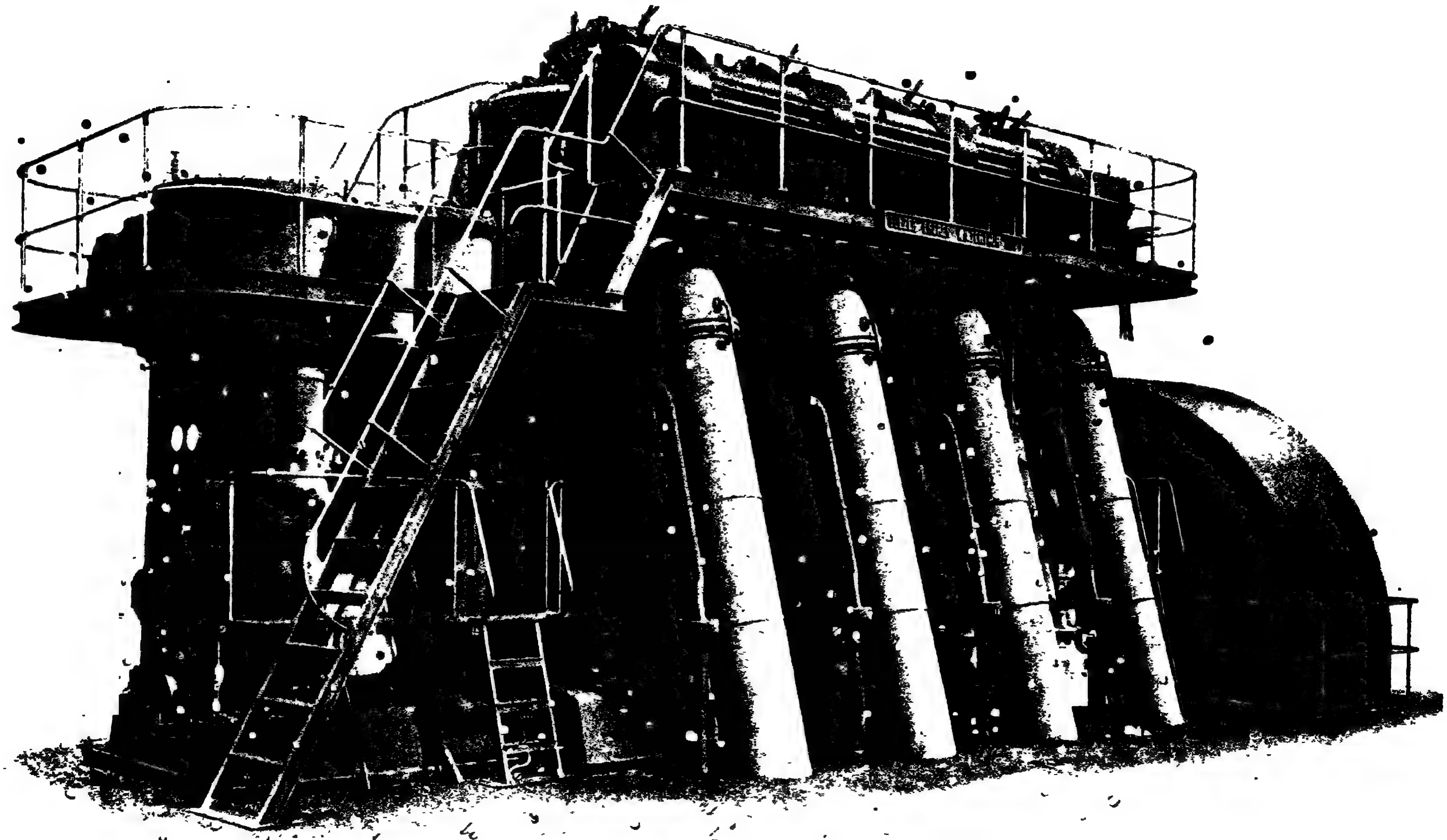

Fig. 202.—Sulzer Stationary 2400-B.H.P. Engine

directly from the governor. The control of the fuel valve is effected by shifting the position of a system of levers. This requires an appreciable amount of power, and would seriously handicap the sensitive action of the governor if operated directly from it. To avoid this a small relay cylinder is used, actuated by compressed air supplied from the low-pressure stage of the compressor. In other respects this engine calls for no particular comment, its mechanical features follow those of the usual four-cycle type.

A 4000-horse-power Engine.—The engine illustrated in fig. 203 has been built by Messrs. Sulzer Brothers, and represents probably the largest Diesel engine ever built commercially. It develops 4000 B.H.P. when running at a speed of 132 R.P.M., and has six cylinders, each 760-mm., 30-in. bore, by 1000-mm., 39·4-in. stroke. The piston speed at normal revolutions is 866 ft. per minute. This engine differs from the one previously described, in that bottom scavenging is employed, and also, although single-acting, separate cross-heads are provided, thus relieving the cylinder liner of all thrust, and materially reducing the piston friction, but at the expense of increased height, and, therefore, weight and cost. The cylinder-heads contain only the fuel and air-starting valves, and are, in consequence, extremely simple castings. The arrangements for scavenging are particularly interesting. Two rows of inlet ports are provided, one immediately above the other, the top of the upper row of ports being slightly above the exhaust, so that they are uncovered first and closed last. The lower series of ports are in open communication with the air inlet manifold, but the upper series are masked by means of either a piston valve or a double-beat poppet valve, until the exhaust ports have been uncovered and air commences to enter through the lower series of ports. As soon as the lower series have been uncovered the piston valve unmasks the upper series. Air then enters through both sets of ports, and continues to do so until the upper set has been covered by the piston on its upward stroke. This is a distinctly neat and thoroughly mechanical method of applying the delaying-valve principle. The pistons have plain concave tops, and the incoming air is given an upward direction by the inclination of the ports, which are so formed as to project the air upwards towards the cylinder-head.

The efficiency of this method of scavenging is well illustrated by the fact that the engine can normally carry a mean effective pressure of about 100 lb. per square inch, with a piston speed of

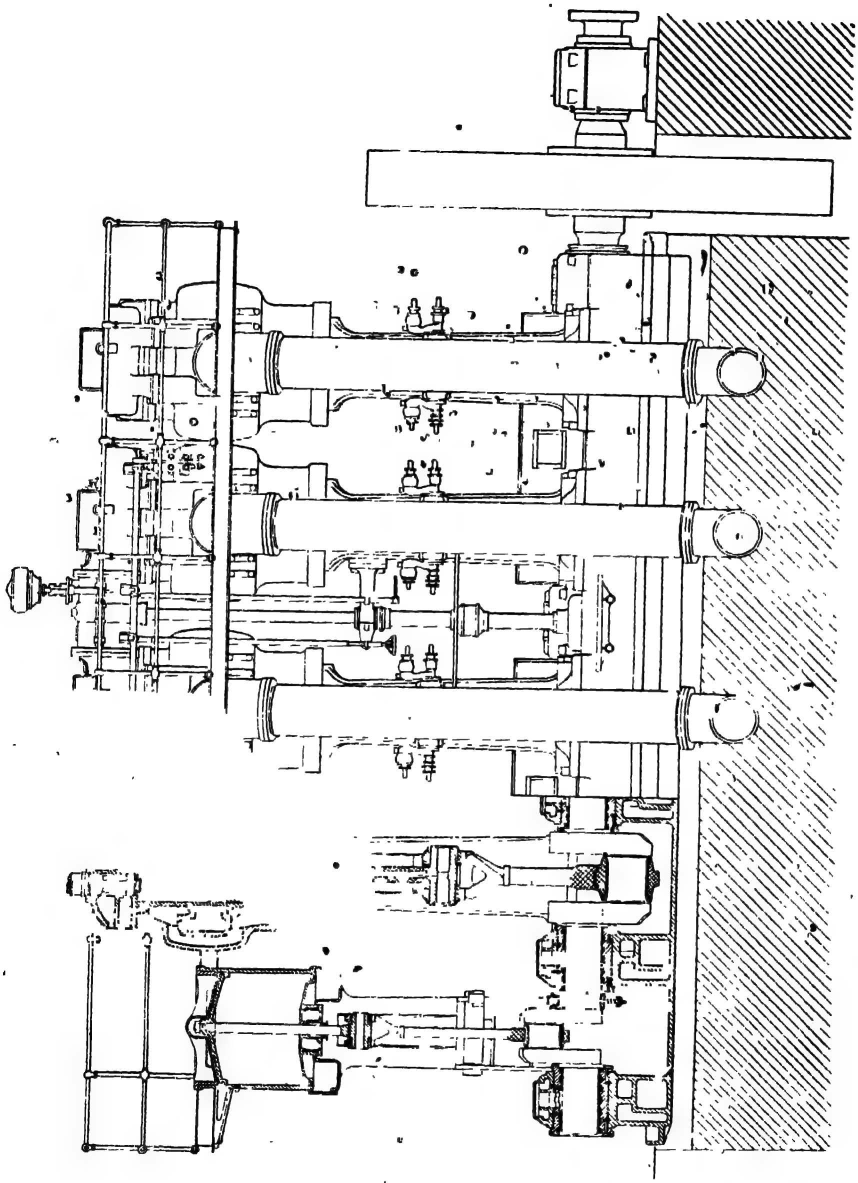

Fig. 204.—Carels-Diesel Stationary 1000-B.H.P. Engine

in which they are made a close fit. For the water-joint between the liners and the exhaust belts a press fit with red lead is relied upon.

The camshaft, it will be noticed, is driven from the centre, which is a desirable feature in a multi-cylinder engine of this size, for, even with only four cylinders, the accumulated "spring" of the

crank- and camshafts is considerable, and results in unequal timing of the cylinders at the free end of the camshaft. The fuel valves employed are very similar to those in the Mirrlees engine already described, and do not call for any particular comment, but it will be noticed that the pistons are very much hollowed in the centre, forming practically a hemispherical combustion chamber.

The high-pressure blast-air for this engine is supplied from an entirely independent source.

The author has been unable to obtain any figures as to the actual results obtained, but it seems probable that the net efficiency will be somewhat low, owing to the small size of the scavenge valves and the large fluid losses which they entail.

Allgemeine Double-piston Engine.—The engines built by the Allgemeine Elektricitäts Gesellschaft, of Berlin, are particularly interesting in that they represent a definite departure from the more orthodox type. In general, they follow the Junkers or Oechelhauser principle, in which two pistons are employed moving in opposite directions, and coupled to cranks at 180 degrees to one another. One piston controls the exhaust and the other the inlet ports, these [illegible] situated at opposite ends of the cylinder. In order t[illegible] of the use of large inlet ports, and also to keep th[illegible] until considerably after the exhaust ports are closed, and [illegible]ect a certain amount of supercharging, the two pistons are [illegible]led to cranks, placed at 165 degrees to one another, so that the upper pistons controlling the exhaust ports have a permanent lead of 15 degrees over the lower ones, in much the same manner as in the Duplex engine, described previously.

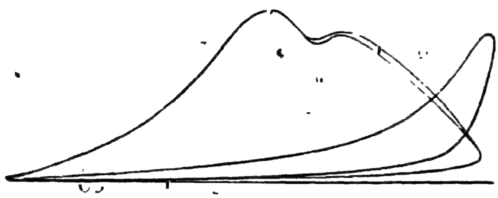

Fig. 205.—Diagrams from Allgemeine Engine

Thanks to the very effective scavenging provided for, and to the super-charging, it is possible to work with an exceptionally high mean effective pressure, and, at the same time, to run at a high rotative speed. In practice, a mean pressure of about 140 lb. per square inch is employed, and that at a rotative speed of no less than 450 R.P.M., in a two-cylinder engine of 250 B.H.P. The combination of this remarkably high mean pressure and high speed of rotation has resulted in the production of an exceptionally light and compact engine. Fig. 205 shows two typical diagrams from one

of these engines, one being taken with the indicator set 90 degrees out of phase. As might be expected, the mechanical efficiency is somewhat low, owing to the high speed, and is said to be in the neighbourhood of 66 per cent. Both pistons are cooled, the lower by means of lubricating oil, which is fed to them from the ordinary lubricating system, and the upper by water, supplied through telescopic pipes. The long return connecting-rods each consist of two steel tubes, and are remarkably light. The upper pistons are provided with cross-heads sliding in guides, consisting of slots formed in the cylinder walls. One interesting feature of these engines is that the pistons are fitted with wide brass bands or rings provided with a number of oil-grooves. These rings serve to distribute the lubricant, and, by closely fitting the cylinder, they maintain the oil-film between the piston and cylinder walls at all times, and so prevent it being broken down by the escape of gases past the main piston-rings.

CHAPTER XXXI

MARINE DIESEL ENGINES

The employment of the Diesel engine for the propulsion of ships, and especially of cargo vessels, has many obvious advantages.

1. The consumption of fuel is approximately one-fourth that of a steam vessel using coal. Consequently, for a given length of voyage only one-quarter of the weight of fuel need be carried; also, the fuel, being liquid, can be stored in the double bottom, and the cargo capacity increased in consequence.

2. Although in this country the cost of fuel oil is so high that the actual cost of fuel is little or no lower than that of steam-engines using coal, the Diesel-engined boat has, by reason of its low fuel consumption, a very large radius of action, which enables it to take its supply of fuel oil at places where it can be obtained at a low price, and for this reason the economy is very marked.

3. The space occupied by the engine is less than that of the engines and boilers of a steam-boat; hence the carrying capacity and earning power are increased.

4. Although the engines themselves at present require more careful supervision than steam-engines, no firemen are required, and the total engineering staff can be reduced.

5. With the further development of the Diesel engine, there is every reason to hope for a considerable saving in weight over the steam plant. At the present time, however, this saving is small, because the marine Diesel engines now built have been somewhat experimental, and very much heavier than is necessary, in order to secure reliability of operation. There is little doubt, however, that the possibilities of weight reduction are greater in the Diesel than in the steam-engine.

6. In the Diesel engine plant there is less risk of fire, which is an important consideration in the case of oil-tank vessels and other vessels carrying inflammable cargo.

These advantages are at present off-set by

1. The greater initial cost of the Diesel engine plant; there is, however, every reason to suppose that this will be reduced considerably when the marine Diesel engine has become more standardized.

2. The marine Diesel engine has not the same ability for manœuvring as the steam-engine, and cannot be run at the very low speeds necessary when coming alongside a dock.

3. The Diesel engine has not, as yet, been able to show anything like the consistent reliability of the steam-engine.

Of these disadvantages none but the last are very serious. Reliability of operation is, obviously, a paramount necessity in any marine engine, and it is in this direction that nearly all the Diesel-driven vessels at present have more or less failed. The causes of failure are various, but are mainly due to

1. Cracked cylinder-heads.
2. Cracked pistons.
3. Trouble with piston-cooling, involving failure of the water circulation, or leakage.
4. Air-compressor troubles.
5. Failure of exhaust valves.

Of these, the first four are common to both the two- and four-cycle type, while the fifth is peculiar to the four-cycle, and no corresponding trouble has been traced to the exhaust ports.

Cracked cylinder-heads have been of more frequent occurrence with two than with four-cycle engines; but this is as might be expected, for the majority of two-cycle engines now in operation on shipboard have four scavenge valves in each cylinder-head. This means six valves in all, which involves a complicated casting, thus greatly reducing the strength of the cylinder-head and interfering with the water circulation.

Piston troubles have been all too frequent, and are generally traceable to a temporary failure of the water circulation or to excessive incrustation with salts from the water. Indirect trouble has also occurred from the same source, due to the leakage of water, especially when telescopic pipes are employed. These pipes must necessarily be fitted in such a position that any leakage from them is liable to find its way into the crank-pit and interfere with the lubrication of the bearings generally. It is understood that several failures of bearings at sea have been directly attributable to this cause. Water-cooling of pistons has always been a source of trouble

and anxiety in all large internal-combustion engines, and is especially objectionable in vertical engines, in which it is difficult to prevent the leakage from reaching the crank-pit, while it is often impossible to stop and repack the glands for long periods. In some cases oil-cooling has been employed in preference to water-cooling. This certainly eliminates the danger of leakage, but, as already pointed out, ordinary lubricating oil is a poor conductor of heat, and it is a difficult matter to withdraw the heat from the pistons, in the first instance, and from the oil afterwards.

Air-compressor troubles are more or less common to all Diesel engines. The compression of air to a pressure of 900 lb. per square inch or more is always a difficult operation, involving considerable risk, both from leakage and from the high temperatures produced. The latter is liable to cause carbonization of the oil and sticking of the valves, and is also a source of real danger from explosion, owing to the ignition of the lubricating oil in the highly heated and compressed air. This is guarded against by curtailing the lubrication of the compressor as far as possible, and by frequently blowing out the air-storage bottles to drain off any oil which may have accumulated. It is earnestly to be hoped that the advances which have recently been made in the production of mechanical fuel-pulverizers will render the air-compressor, with all its attendant troubles, unnecessary.

At the present time both two- and four-cycle engines are being employed, and it is worthy of note that, up to the present, the four-cycle engine has met with the greater measure of success. The relative advantages of the two types have already been discussed. In spite of the greater success of the four-cycle type, there seems to be little doubt that the two-cycle is best adapted for marine work, on the score of the greater power obtainable from a given size and weight of engine, and also owing to the lower speed at which two-cycle engines can be run with advantage. The absence of exhaust valves and the simplicity of reversing are both strong points in favour of the two-cycle type. The poor success of the two-cycle marine engine is probably to be attributed rather to the faulty application of this cycle than to any disability of the cycle itself.

One of the results of the adaptation of the Diesel engine to marine practice has been that the trunk piston of the ordinary single-acting type is now replaced by a piston-rod and cross-head. This change was, no doubt, originally brought about by the desire

of the makers to conform as far as possible to steam-engine practice, and so anticipate and avoid the prejudice which marine engineers would have against the trunk piston. The change has been altogether desirable, for not only has the use of an external cross-head improved the mechanical efficiency and reduced the wear on the liner, but it also enables the piston to be made with a considerable amount of clearance, and thereby eliminates risk of seizure due to deformation. So successful has the use of an external cross-head been found that it has now been adopted, in many cases, for stationary engines also.

Another result of the adaptation of the Diesel engine to marine work has been the development of the open type of engine in preference to the enclosed crankcase with forced lubrication. This change, which has also been made to meet the prejudice of marine engineers, has been most undesirable, and has led to trouble with bearings of a kind that seldom, or never, occurs in stationary engines, and it is worthy of note that most of the successful Diesel-engine installations on board ship have had enclosed crankcases and forced lubrication, in defiance of the steam-engineers' prejudices.

Burmeister & Wain Engine.—Credit for having the enterprise first to install a Diesel engine in a large sea-going vessel is probably due to Messrs. Burmeister & Wain, of Copenhagen, who both built and engined the *Selandia*, a vessel of about 4963 tons, in 1912. This vessel is equipped with two eight-cylinder four-cycle Diesel engines, driving twin screws, and developing 2250 aggregate horse-power. The advent of the *Selandia* was soon followed by a number of other vessels built by the same firm, of which the largest is the *Fionia*, of 5219 tons, and equipped with two six-cylinder engines, developing collectively 4000 I.H.P. An illustration of one of the *Fionia's* engines is given in fig. 206. The diameter of cylinders is 29·2 in., the stroke 43·5 in., and the normal speed 100 R.P.M. Unlike most of the marine Diesel engines, Messrs. Burmeister & Wain employ a totally enclosed crank-chamber, with forced lubrication to all the main bearings. Separate cross-heads are employed for the pistons, as is now usual in marine-engine practice, and it is noticeable that very little effort has been made to reduce the height of the engine. The pistons are cooled with oil from the lubricating system. The valves are all operated from a single horizontal camshaft carried along the top of the base-chamber, just below the foot of the cylinders. All valves,

Fig. 206.—Burmeister & Wain Engine

including the fuel valves, open inwards, the latter being a somewhat unusual arrangement in a Diesel engine, but Messrs. Burmeister & Wain have employed this type of fuel valve for some years.

The construction of the exhaust valves is somewhat unusual. Both the head and stem are of steel, but for the seating itself an annular ring of cast iron is employed. It is claimed that by adopting this construction the valves will remain tight for a longer period without requiring grinding-in, while the danger of the head and stem parting company is reduced. The camshaft is centrally driven by means of a long train of spur-wheels connecting the crank- and camshafts. This has been found considerably more reliable than spiral gearing, and is undoubtedly an excellent arrangement, provided that the wheels are sufficiently accurately machined and spaced to ensure silent running, which is no easy problem. Reversing is effected by sliding the camshaft bodily along. To accomplish this a compressed-air servo-motor is employed, driven by air supplied from the intermediate-stage compressors. Before sliding the camshaft it is necessary first to raise all the valves, an operation which is carried out automatically.

The high-pressure blast air is supplied from two sources. The two low-pressure stages of the compressors are driven by auxiliary Diesel engines, by means of which the air is compressed to a pressure of 300 lb. per square inch, and delivered to large storage tanks. Thence it is distributed for operating certain auxiliary machinery, and also for starting and manœuvring the main engines. For the high-pressure air a small single-stage compressor is fitted to the forward end of the main engine. This pump draws air from the main storage tanks at a pressure of 300 lb. per square inch, and delivers it to small steel air-bottles at a pressure of about 900 lb. per square inch. The remaining mechanical details of these engines do not call for any particular comment. The construction throughout is exceptionally massive, and it is evident that reliability of operation has been the primary object, and that weight and cost have been regarded as purely secondary considerations.

The following test figures have been kindly supplied by the makers, who state that they were obtained from three different vessels at sea:—

M.S. *Siam* (two 8-cylinder engines, 23·2 in. × 31·6 in.)

	Port Engine.	Starboard Engine.	
R.P.M.	123·5	123·5	—
I.H.P.	1514	1527	
Collective I.H.P.	—	—	= 3041
M.E.P. (pounds per square inch)	89·5	90·3	—
Oil (pounds per I.H.P. hour)	—	—	= 0·30
Calorific value of oil (B.T.U.s per pound)	—	—	= 17800
Indicated thermal efficiency	—	—	= 47·6 %

M.S. *Pedro Christoffersen* (two 8-cylinder engines, 19·7 in. × 26 in.)

	Port Engine.	Starboard Engine.	
R.P.M.	145·4	145·2	—
I.H.P.	887·9	851·6	
Collective I.H.P.	—	—	= 1739·5
M.E.P. (pounds per square inch)	75·5	72·4	—
Oil (pounds per I.H.P. hour)	—	—	= 0·323
Calorific value of oil (B.T.U.s per pound)	—	—	= 17800
Indicated thermal efficiency	—	—	= 44 %

M.S. *Annam* (two 8-cylinder engines, 23·2 in. × 31·6 in.)

	Port Engine	Starboard Engine	
R.P.M.	137·4	139	—
I.H.P.	1750·7	1773·9	—
Collective I.H.P.	—	—	= 3524·6
M.E.P. (pounds per square inch)	93	93	—
Oil (pounds per I.H.P. hour)	—	—	= 0·327
Calorific value of oil (B.T.U.s per pound)	—	—	= 17800
Indicated thermal efficiency	—		= 43·8 %

Indicator diagrams taken from the M.S. *Siam* engines during the above test are shown in fig. 207. It will be noticed that the maximum pressure rises considerably above the compression pressure in all cases, and it is no doubt to this fact that the very high efficiency of this engine is largely to be attributed. The mean effective pressure in each instance is low for a four-cycle engine, but this has evidently been kept as low as is commercially possible, with a view to reducing the risks from heat stress in the cylinder-heads and pistons and to protect the exhaust valves.

The mechanical efficiency of the *Siam's* engines is stated to be 85 per cent, but this figure includes only the loss in the high-pressure stages of the blast-air compressors. It is, nevertheless, a very high figure, but quite a possible one in engines of this size, using separate cross-heads, and running at a piston speed of only 650 ft. per minute. If the power absorbed by the low-pressure stages of the blast-air compressors be taken into account, the over-

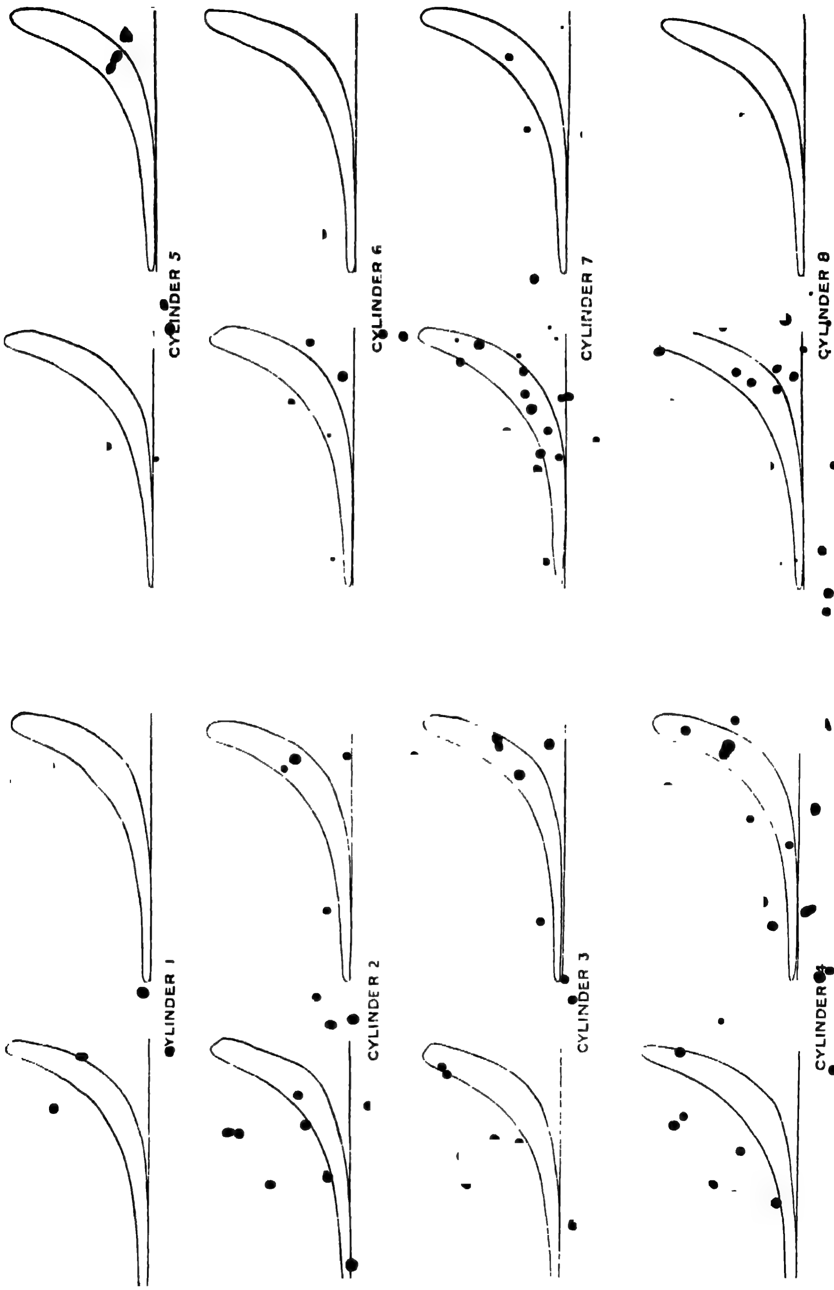

Fig. 207.—Burmeister & Wain Engine Diagrams

all mechanical efficiency then becomes 75 per cent; but this, of course, includes all the losses in the auxiliary Diesel engines driving the compressors. If the compressors were driven directly by the main engines it is probable that the mechanical efficiency would

then be about 78 per cent, and the brake thermal efficiency 37 per cent, which is a remarkably high figure.

Werkspoor Marine Engine.—The Werkspoor engine, made

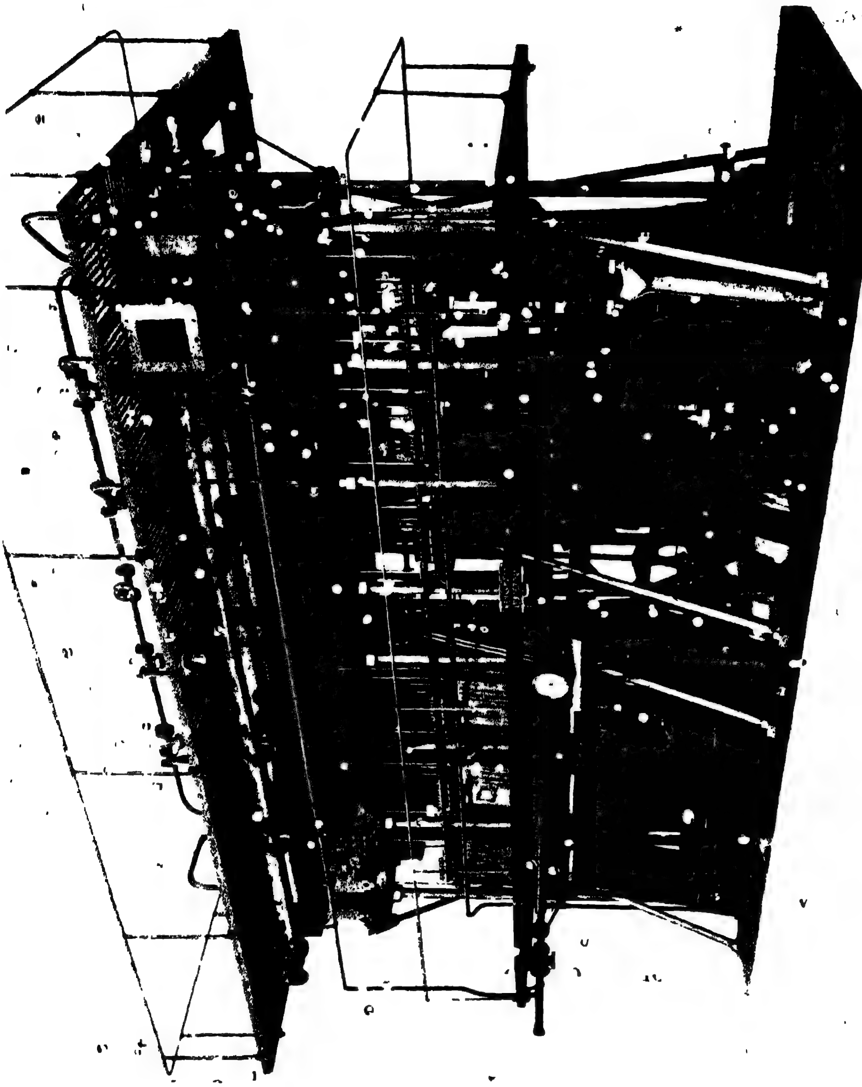

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Fig. 208.—Diesel Engine installed on M.S. *Juno*

by the Nederlandsche Fabriek of Amsterdam, is another example of a successful four-cycle marine engine. This engine, which is shown in fig. 208, differs from most others in that the usual heavy cast-iron frames are replaced by light steel columns, and diagonal tie-rods are employed to give the necessary rigidity. The engine

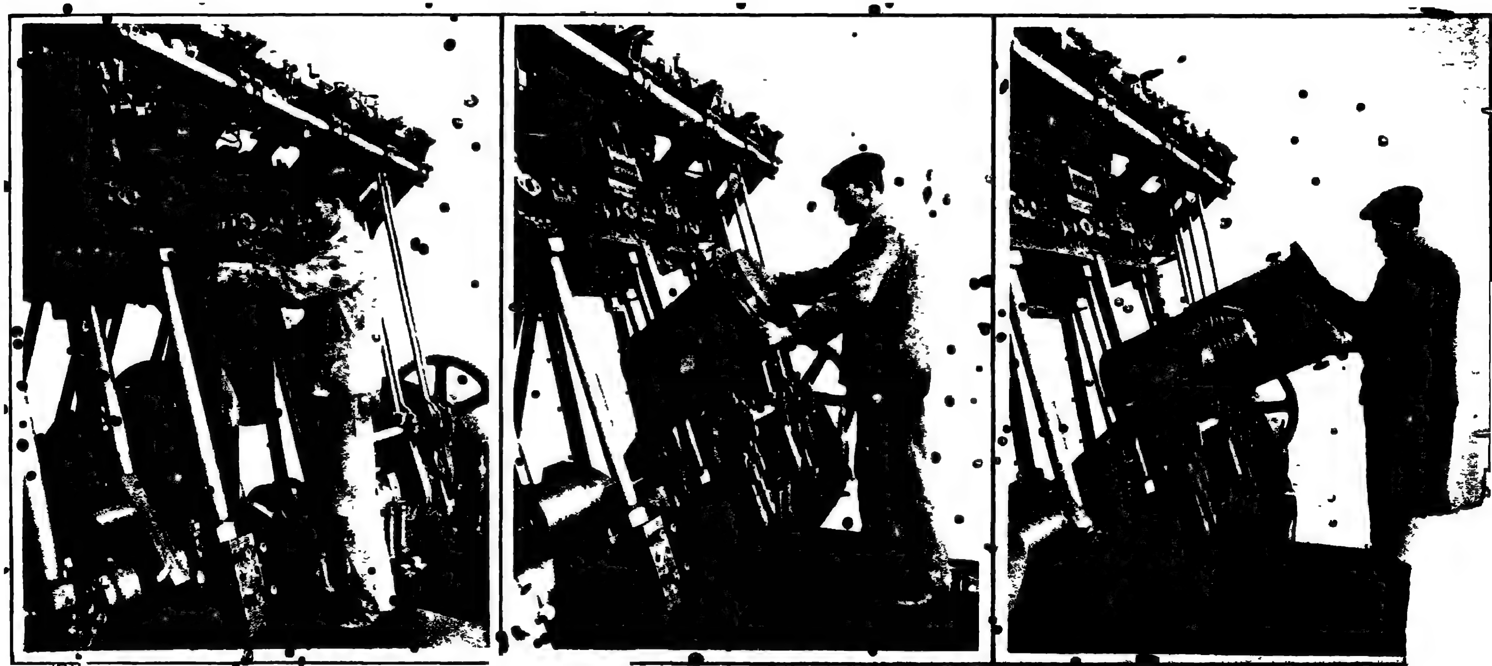

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Fig. 212.—Three Views of Inspecting Pistons on "Werkspoor" Gunboat Diesel Engine

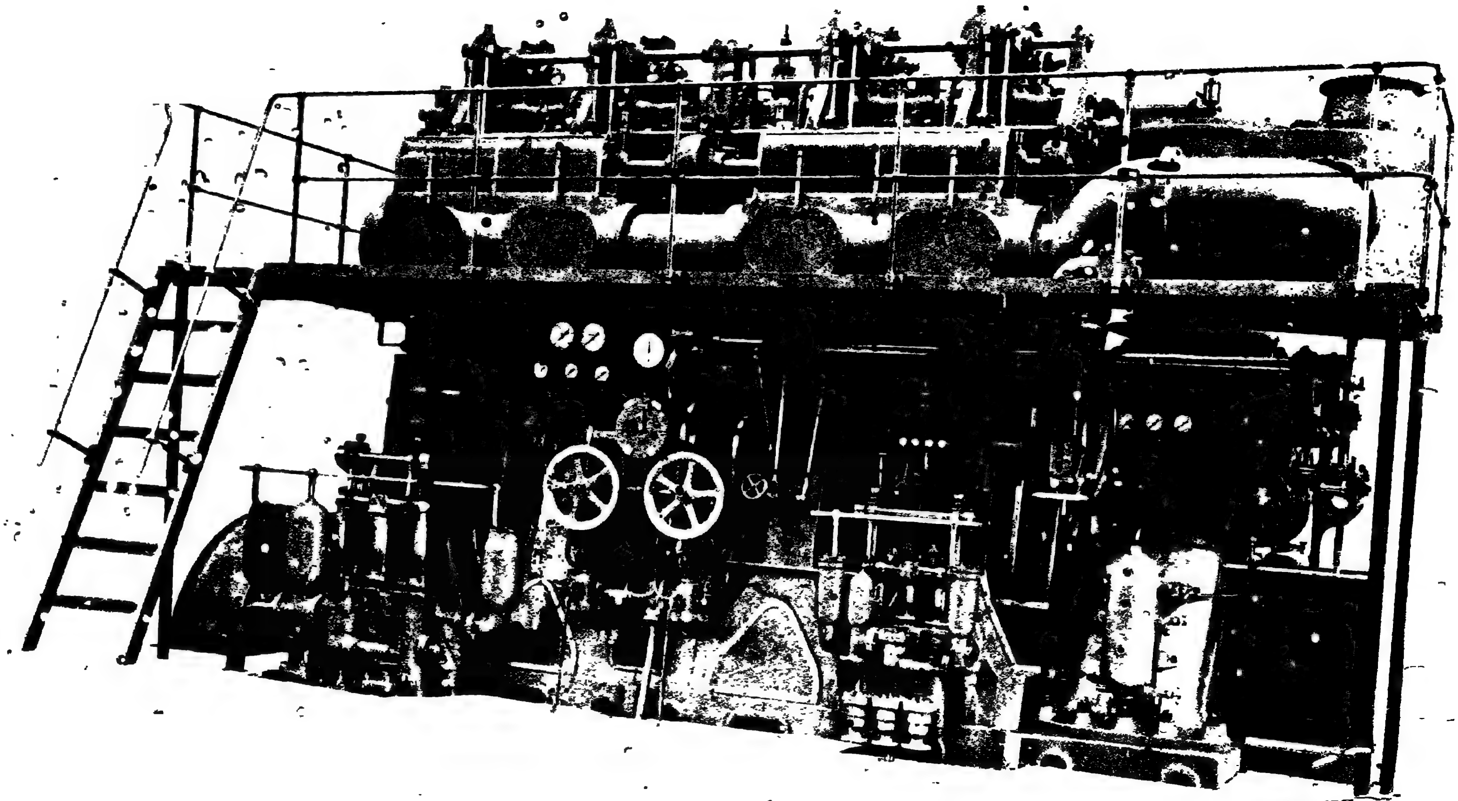

Fig. 213.—Sulzer Marine E[illegible]

upper part of the liners are cast in one piece, an arrangement which obviates the great thickness of uncooled metal at the point where the liner comes into contact with the cylinder-head; but it is open to the objection that the metal most suitable for the cylinder-head

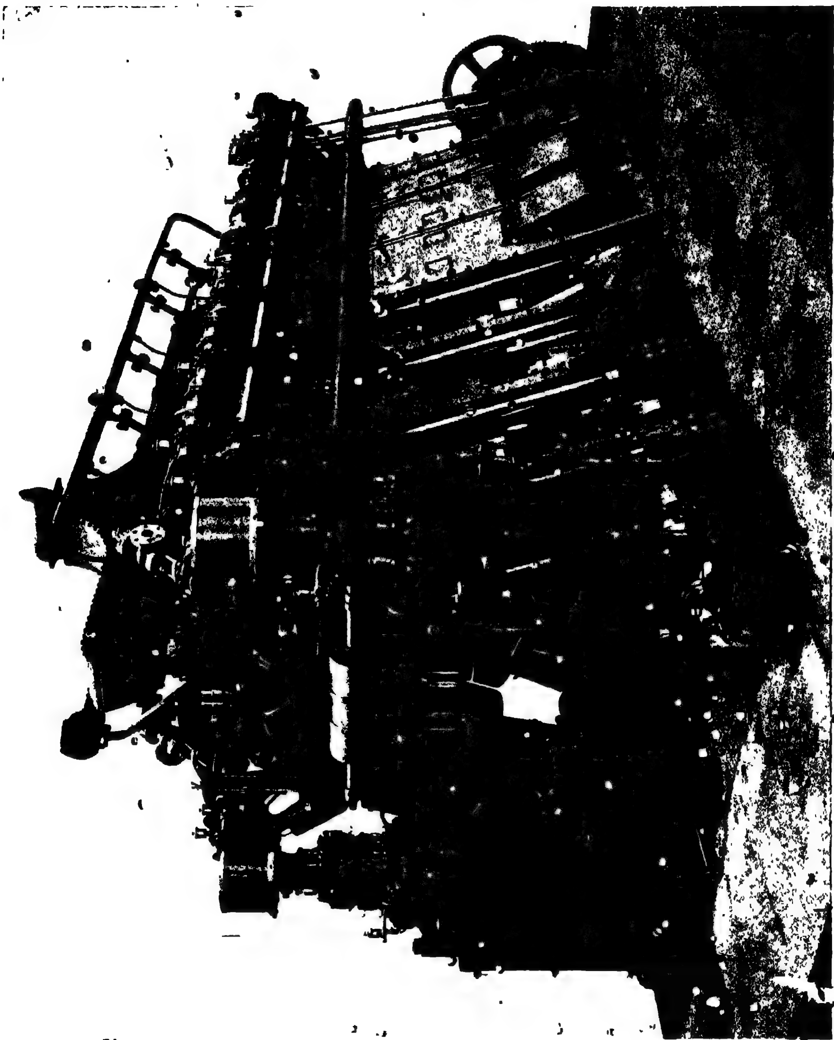

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Fig. 211.—Twin Screw "Werkspoor" Diesel Engines of 1200 B.H.P. for Shallow-draft Gunboat

(i.e. a metal which will flow freely in the mould and will not b subject to severe contraction stresses) has not necessarily the bes wearing qualities, and therefore may not be very suitable as cylinder liner.

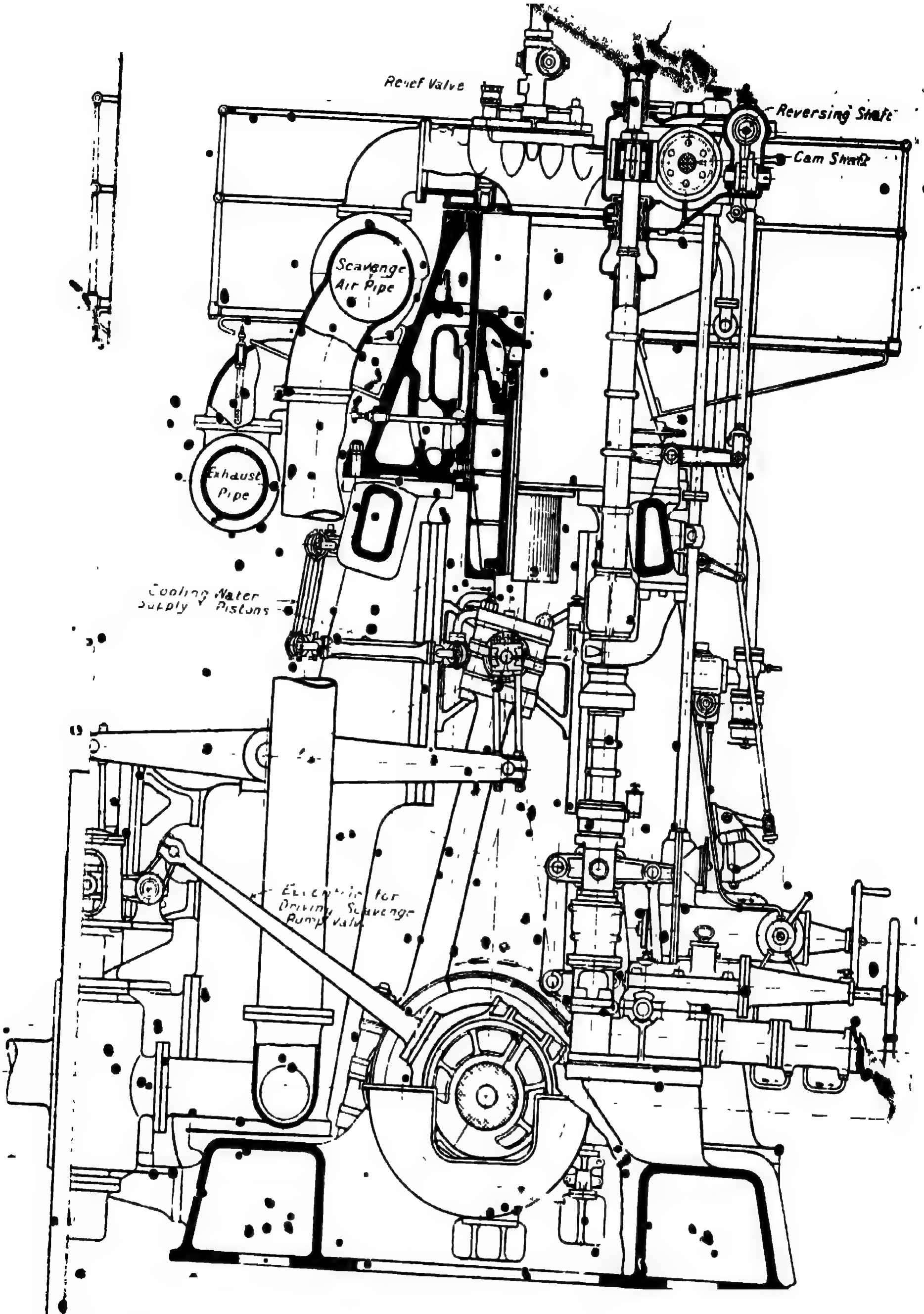
The high-pressure air-compressors are driven from the cros heads. The low-pressure stage is entirely independent, but t

intermediate and high-pressure stages are in tandem. In most respects the engine follows the usual Diesel practice, but the extreme accessibility of the pistons and the method of driving the camshafts by means of direct cranks in place of gearing are both very attractive features, while the general construction of the engine is such that it should be considerably lighter than most other marine Diesel engines. Werkspoor engines have now been in successful operation in a number of boats, and are reported to have given excellent results.

The Nederlandsche Fabrik have also built a most interesting pair of engines for a small Dutch gunboat. These engines, which are illustrated in figs. 211 and 212, have inclined cylinders, and are said to be exceedingly light for the power developed, as indeed they appear to be.

Sulzer Two-cycle Marine Engine.—Of the two-cycle marine engines, the Sulzer Diesel has been probably the most successful during the last few years. One of these engines, developing about 850 to 1000 horse-power, has been fitted in the motor-ship *Monte Penedo*, and up to the time of writing has run with great regularity. This engine, which is illustrated in figs. 213 and 214, has four cylinders, each 18·5-in. bore by 26·8-in. stroke, and develops 850 B.H.P. at a speed of 150 R.P.M., corresponding to a piston speed of 675 ft. per minute. To develop 850 B.H.P. at this speed, and with these cylinder dimensions, requires a brake mean pressure of no less than 79 lb. per square inch; and, since the mechanical efficiency cannot be higher than about 72 per cent, the indicated mean pressure must be approximately 110 lb. per square inch. This is an extremely high figure for a two-cycle engine, and proves that the system of port or bottom scavenging employed can be reasonably efficient. Moreover, the marked success which this engine has achieved shows that a plain combustion chamber, free from large valves, can safely withstand high mean pressures.

The mechanical details of this engine are practically identical with those of the 4000-B.H.P. stationary Diesel already described, and the same system of scavenging—with two rows of inlet ports and a delaying valve—is adopted. The performance of this engine is decidedly interesting, because it shows that if bottom or port scavenging be employed, and be correctly designed, it is possible to obtain as much as double the power from a two-cycle as from a four-cycle engine of given cylinder dimensions and speed. Another feature of this engine is that, in common with the Burmeister &


Relief Valve
Reversing Shaft
Cam Shaft
Scavenge Air Pipe
Exhaust Pipe
Cooling Water Supply to Pistons
Eccentric for Driving Scavenge Pump Valve

EAVESTONE

h engines, an enclosed crank-chamber is employed with forced plication. It is contrary to marine-engine practice, but in both ps it appears to have been justified by entire immunity from uble with bearings, an immunity which has not been shared by er types of marine Diesel engines.

Another excellent feature of this engine is that all four cylinders ve open, flat sides, and are bolted together to form one single

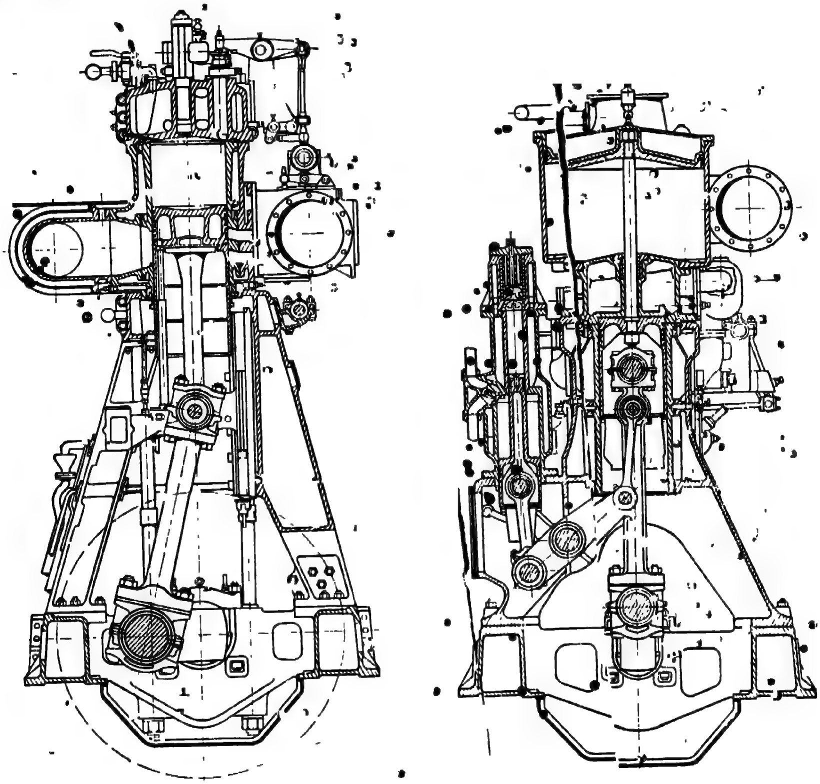


Fig. 214.—Sulzer Marine Engine

block, thus ensuring great rigidity. The cast-iron standards are relieved from tension by means of long steel columns, carried right up through the cylinder-heads, which take the greater part of the tensile stress, and thus allow of considerable reduction in the weight of the engine.

Reversing, in an engine of this type, is a somewhat complicated matter, and as far as the fuel valves are concerned it is effected in this case by rotating the cam shaft through a small angle. This is accomplished by merely raising the vertical shaft which connects

the crankshaft and the camshaft by spiral gearing. This sha[ft is] formed in two parts, with a telescopic joint or sleeve between t[hem,] so that one half can be raised together with the spiral gear wh[ich] which gives the necessary rotation. For the reversal of the starting valves two separate cams are employed for each val[ve,] and, since the camshaft has no longitudinal movement, this chang[e-]over is effected by moving the roller instead. Finally, the delayin[g] valve requires reversing, and this is accomplished by means of [a] simple link gear The whole arrangement appears at first sigh[t] to be very complicated, but in practice it is not so The piston[s] in this engine are water-cooled by means of telescopic pipes placed within the crank chamber, and these apperently have not given rise to any trouble from leakage. The fuel consumption of this engine is said to be, approximately, 0.46 lb. per B.H.P.-hour when running on full load and driving all the necessary auxiliaries The heating value of the fuel-oil is not given, so that it is not possible to make any comparisons with the four-cycle types.

The Carels Engine.—The Carels-Diesel engine has been employed in a number of large motor vessels, and is probably the most widely used of all two-cycle marine Diesel engines. This also is a single-acting cross-head engine, but is of the open type. In general, this engine does not differ materially from the 1000-B.H.P. stationary engine already described. The principal differences are

1. That the cylinders are formed separately from the **A** frames to which they are bolted.

2. The scavenge pumps are driven by rocker arms from the main cross-heads, and are fitted with automatic plate valves in preference to piston valves. This is a desirable feature, in that it facilitates reversing, and reduces the fluid losses when the engine is running at low speeds.

3. The separate **A** brackets are all bolted together at the base of the cylinders, thus greatly increasing the rigidity of the engine as a whole.

Reversing is accomplished, in so far as the scavenge valves are concerned, by raising or lowering the vertical shaft, and so rotating the camshaft through a small angle relative to the cranksnaft. The fuel valves are reversed by means of two cams, and since the camshaft has no longitudinal movement, it is necessary to move the rollers of the rocker arms sideways in order to bring them into

with the second set of cams. All the rollers are kept in position by separate horizontal shafts, which can be moved longitudinally in order to change over from one set of cams to the other.

From the sectional drawings (fig. 215) it will be seen that the lower ends of the cylinder liners are provided with stuffing-glands, whose function it is to retain the lubricating oil. The high-pressure blast-air compressor is driven from the forward cross-head through the medium of a rocking beam, in the same manner as the scavenge pumps. The cylinder dimensions of the 1800-B.H.P. six-cylinder Carels marine engine, as fitted in a large oil-tank vessel for the British Admiralty, are: bore, 600 mm., or 23·6 in., with a stroke of 1100 mm., or 43·4 in., and the engine runs normally at a speed of 100 R.P.M., corresponding to a piston speed of 716 ft. per minute. At the rated load of 1800 B.H.P. the brake mean pressure is 63 lb. per square inch, and if the mechanical efficiency be taken as 70 per cent, the indicated mean pressure is 90 lb. per square inch. This is a somewhat low pressure for a Diesel engine, but is doubtless necessitated by the complicated form of the combustion-head and the extremely severe stresses set up in it, due to temperature differences if higher mean pressures be employed.

In one vessel, the *Earestone*, fitted with a four-cylinder Carels-Diesel engine of 800 horse-power, mean pressures as high as 125 lb. per square inch were attempted, but it is reported that these high pressures led to great trouble with cracked combustion-heads, and had to be reduced. This particular engine ran at the low speed of 90 R.P.M., so that the heat flow, which is nearly proportional to the speed, would not be very great in this case, and not nearly so great as in the Sulzer-Diesel engine of the *Monte Penedo*.

In the earlier types of Carels marine engines cast steel was employed for the combustion-heads, and this, at first, was perfectly satisfactory; but in the later engines small cracks developed in the combustion-heads.

Cast iron is now employed, and has been found satisfactory, but there is some evidence to indicate that for very large engines steel would be preferable, on account of its greater tensile strength, provided that the initial casting stresses can be removed by annealing and suitable heat treatment. A modification in the cylinder design has been introduced by the Reiherstieg Company, one of Messrs. Carels' licencees, whereby the cylinder liner is provided with a wide flange, enabling the water-cooling to be

carried higher up, and thus obviating the great thickness of uncooled metal at the top of the liner.

The fuel consumption of the Carels engine is said to be approximately 0·47 lb. per B.H.P.-hour, of Admiralty fuel-oil having a calorific value of about 18,300 B.T.U.s per pound, corresponding to a brake thermal efficiency of 29·7 per cent.

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